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REPORT ODAI-1270-8-79

HYPERVELOCITY WIND TUNNEL COMPONENTS STRUCTURAL EVALUATION

Prepared for:

NAVAL SURFACE WEAPONS CENTER
WHITE OAK, SILVER SPRING, MARYLAND

By

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241 Curry Hollow Road
Pittsburgh, PA 15236

FINAL REPORT

on

CONTRACT NO. N60921-78-C-0013

Volume 2 of 2

(May 1979)

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FINAL REPORT
HYPERVELOCITY WIND TUNNEL
COMPONENTS

ABSTRACT

A structural evaluation of the large threaded Pressure Vessels in the wind tunnel facility was performed using finite element techniques coupled with fatigue and fracture mechanics analyses of the critical locations. The results of this evaluation show that these threaded pressure vessels have limited fatigue life due to high stress concentrations at the root of the thread root radii in the threaded end closures. Design modifications were made to the most critical end closures (Bottom End of Mach 14/18 Heater Vessel and Inlet End of Driver Vessel) to increase the design life of these pressure vessels. The design life of all of the threaded pressure vessels was also increased by reducing the maximum pressure at which they are operated. Periodic inspection requirements which account for variable pressure cycling and mean stress effects were also developed for the critical areas of these threaded pressure vessels.

The net result of the design modifications, reduced operating pressures and periodic inspection requirements is to increase the design life and confidence in the safety related structural integrity of the threaded pressure vessels in the wind tunnel facility.

APPENDIX 1A

STRUCTURAL EVALUATION OF MANIFOLDS

A 2

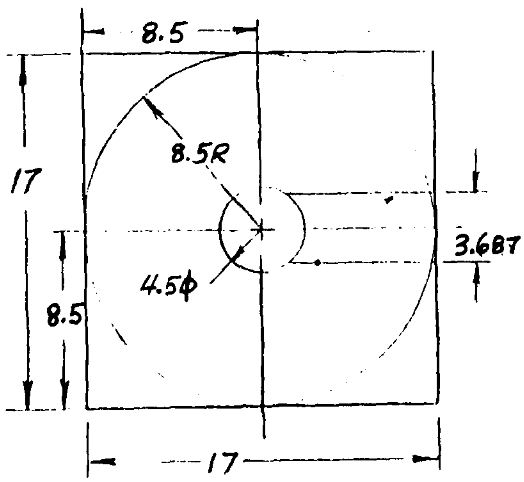
STRUCTURAL EVALUATION OF MANIFOLDS

The Hypersonic Wind Tunnel facility has two manifolds, an inlet and exit manifold. These are used for all three loops, M10, M15 and M18. The inlet and exit manifolds are shown on National Forge Drawings 4-01513, Rev. 0, and 4-01514, Rev. A, respectively. The material specifications for the manifolds are listed below.

Component	Material	σ_u	σ_y
Inlet Body	--	143,000	131,000
Exit Body	--	142,000	129,500
Studs	ASTM A193, GRB-7	125,000	105,000
Flange	AISI 4340	135,000	120,000

INTERSECTION OF CROSS TUNNELS

Exit Manifold



Consider square block as thick-walled cylinder, $R_i = 2.25$, $R_o = 8.5$. At R_i:

$$\sigma_\theta = \frac{P_i(a^2 + b^2)}{b^2 - a^2}$$

$$\sigma_r = P_i$$

$$\sigma_z = \frac{P_i a^2}{b^2 - a^2}$$

For the design pressure of 46,000 psi:

$$\sigma_\theta = \frac{(46{,}000)(2.25^2 + 8.5^2)}{8.5^2 - 2.25^2} = 52{,}932 \text{ psi}$$

$$\sigma_r = -46{,}000 \text{ psi}$$

$$\sigma_z = \frac{(46{,}000)(2.25^2)}{8.5^2 - 2.25^2} = 3{,}466 \text{ psi}$$

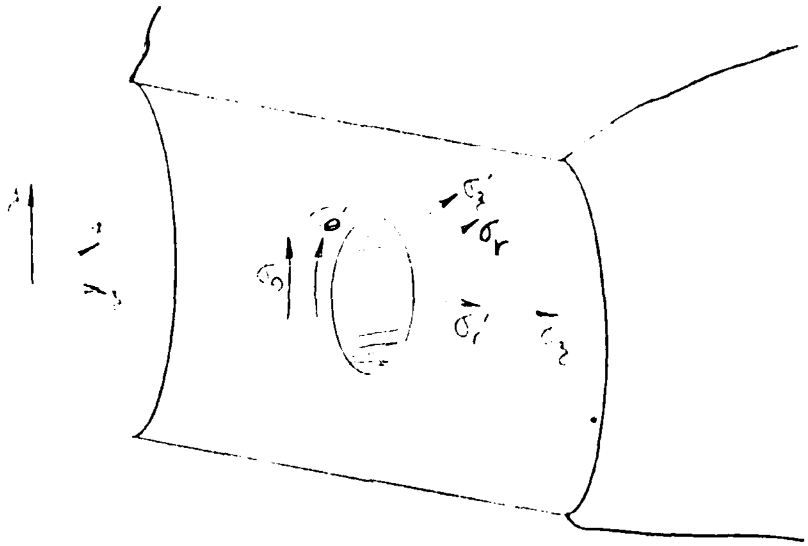
The horizontal 3.6870 hole can also be looked at as if it were a thick-walled cylinder with $R_i = 1.8435$ and $R_o = 8.5$. Therefore, for this "cylinder":

$$\sigma_\theta' = \frac{(46{,}000)(1.8435^2 + 8.5^2)}{8.5^2 - 1.8435^2} = 50{,}541 \text{ psi}$$

$$\sigma_r' = -46{,}000 \text{ psi}$$

$$\sigma_z' = \frac{46{,}000(1.8435^2)}{8.5^2 - 1.8435^2} = 2{,}271 \text{ psi}$$

Now looking at the intersection of the two holes:



$$\sigma_x = (\sigma_z' + \sigma_r) = -48,271 \text{ psi}$$

$$\sigma_y = (\sigma_\theta' + \sigma_\theta) = 103,473 \text{ psi}$$

$$\sigma_z = (\sigma_z + \sigma_r') = 49,466 \text{ psi}$$

This gives a stress intensity of $\sigma_y - \sigma_x$ = 151,744 psi, or 3.3 P_i.

Since the stresses will decrease with decreasing R_i, this is the maximum stress at the intersections in the exit manifold.

Following the procedure outlined in Appendix 3A we can evaluate the fatigue life:

σ_{RANGE} = 151,744 psi; σ_{ALT} = 75,872 psi

σ_{MEAN} = 75,872 psi; σ_y = 120,000 psi; σ_u = 135,000 psi

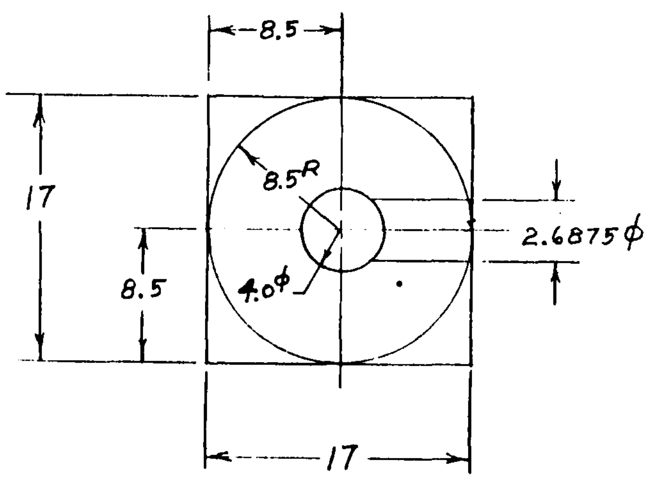
$\sigma'_{MEAN} = \sigma_y - \sigma_{ALT}$ = 120,000 - 75,872 = 44,128 psi

$$\sigma_{eq} = \frac{7\sigma_{ALT}}{8 - \left[1 + \frac{\sigma'_{MEAN}}{\sigma_u}\right]^3} = \frac{(7)(75,872)}{8 - \left[1 + \frac{44,128}{135,000}\right]^3}$$

$$\sigma_{eq} = 93,770 \text{ psi}$$

N = 1,350 cylces(from Figure 3A-7A).

Inlet Manifold



Consider square block as thick-walled cylinder, $R_i = 2.0$, $R_o = 8.5$. At R_i:

$$\sigma_\theta = \frac{P_i(a^2 + b^2)}{b^2 - a^2}$$

$$\sigma_r = P_i$$

$$\sigma_z = \frac{P_i a^2}{b^2 - a^2}$$

For the design pressure of 60,000 psi:

$$\sigma_\theta = \frac{(60,000)(2.0^2 + 8.5^2)}{8.5^2 - 2.0^2} = 67,033 \text{ psi}$$

$$\sigma_r = -60,000 \text{ psi}$$

$$\sigma_z = \frac{(60,000)(2.0^2)}{8.5^2 - 2.0^2} = 3,516 \text{ psi}$$

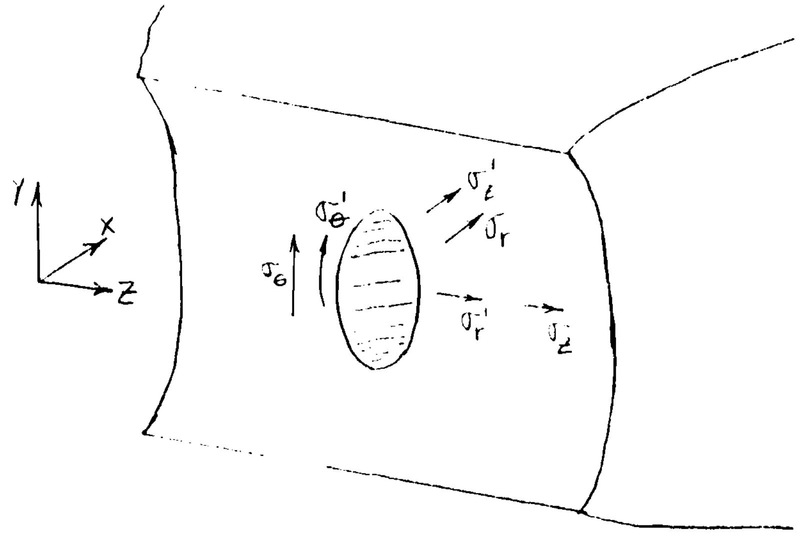
The horizontal 2.6875 hole can also be looked at as if it were a thick-walled cylinder with $R_i = 1.34375$ and $R_o = 8.5$. Therefore, for this "cylinder":

$$\sigma_\theta' = \frac{(60,000)(1.34375^2 + 8.5^2)}{8.5^2 - 1.34375} = 63,076 \text{ psi}$$

$$\sigma_r' = -60,000 \text{ psi}$$

$$\sigma_z' = \frac{60,000(1.34375^2)}{8.5^2 - 1.34375^2} = 1,538 \text{ psi}$$

Now looking at the intersection of the two holes:



$$\sigma_x = (\sigma_z' + \sigma_r) = -61,538 \text{ psi}$$

$$\sigma_y = (\sigma_\theta' + \sigma_\theta) = 130,109 \text{ psi}$$

$$\sigma_z = (\sigma_z + \sigma_r') = 63,516 \text{ psi}$$

This gives a stress intensity of $\sigma_y - \sigma_x$ = 191,647 psi, or 3.194 P_i.

Since the stresses will decrease with decreasing R_i, this is the maximum stress at the intersections in the inlet manifold.

Following the procedure outlined in Appendix 3A we can evaluate the fatigue life: .

σ_{RANGE} = 191,647 psi; σ_{ALT} = 95,824 psi

σ_{MEAN} = 95,824 psi; σ_y = 120,000 psi; σ_u = 135,000 psi

$\sigma'_{mean} = \sigma_y - \sigma_{ALT}$ = 120,000 - 95,824 = 24,176 psi

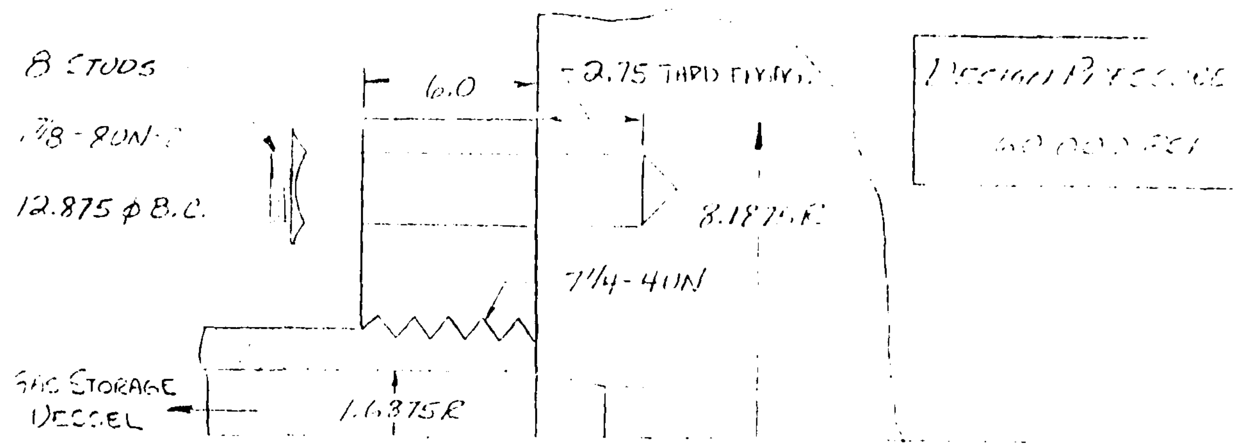
$$\sigma_{eq} = \frac{7\sigma_{ALT}}{8 - \left[1 + \frac{\sigma_{MEAN}}{\sigma_u}\right]^3} = \frac{(7)(95,824)}{8 - \left[1 + \frac{24,176}{135,000}\right]^3}$$

σ_{eq} = 105,543 psi

N = 900 cycles (from Figure 3A-7A).

FLANGE AND FLANGE STUDS INLET MANIFOLD

The details of the inlet flange and studs are shown below:



Pressure load on studs and flange:

$$F = \pi R^2 P = \pi(1.6875)^2(60,000) = 5.368 \times 10^5 \text{ lbs}$$

The tensile area of each stud is - $A_T = 2.401$ in.^{2}, which gives a tensile stress in each stud of:

$$\sigma = \frac{5.368 \times 10^5}{(8)(2.401)} = 27,800 \text{ psi}$$

We also must check the adequacy of the thread engagement length. From NBS Handbook H-28, the length of thread engagement required is:

$$L_e = \frac{2 \times \text{MAX } A_s}{S_s \text{MIN}}$$

where: A_s = maximum stress area (external thread)

S_s = area in shear of external thread

Following the procedure outlined in Appendix A5 of NBS Handbook H28 (1969):

$$A_s = 0.5(C_1 K_n \text{min} \times \frac{L_e}{D} \times D_s \text{ max})$$

$\frac{L_e}{D}$ from Figure A5.3 for 1-7/8 dia. = 0.6255

$$(A_s)_{max} = 0.5\ [(2.356)(1.740)(0.6255)(1.8725)] = 2.401$$

$$S_s = K_n \text{max}(C_1 - C_5\ T_{Kn})$$

$$= 1.765(2.356 - (14.51)(0.03) = 3.390$$

$$\therefore L_e = \frac{(2)(2.401)}{3.390} = 1.416 \text{ in.}$$

This value is less than the specified 2.75 in., and, therefore, the studs have adequate engagement.

Now looking at the 7-1/4 - 4UN thread. This thread size is outside of the range of the NBS Handbook; however, if we look at the ratios of the pitch diameters and lengths of engagement, it can be seen that this joint will have a large shear area than the studs and consequently will be lower in stress:

$$\text{RATIO} = \frac{(7\text{-}1/4)(6)}{(8)(1\text{-}7/8)(2.75)} = 1.05$$

Finally, looking at the flange. Consider the flange to be a circular plate, built-in at its edges, and loaded in a circular region by 60,000 psi.

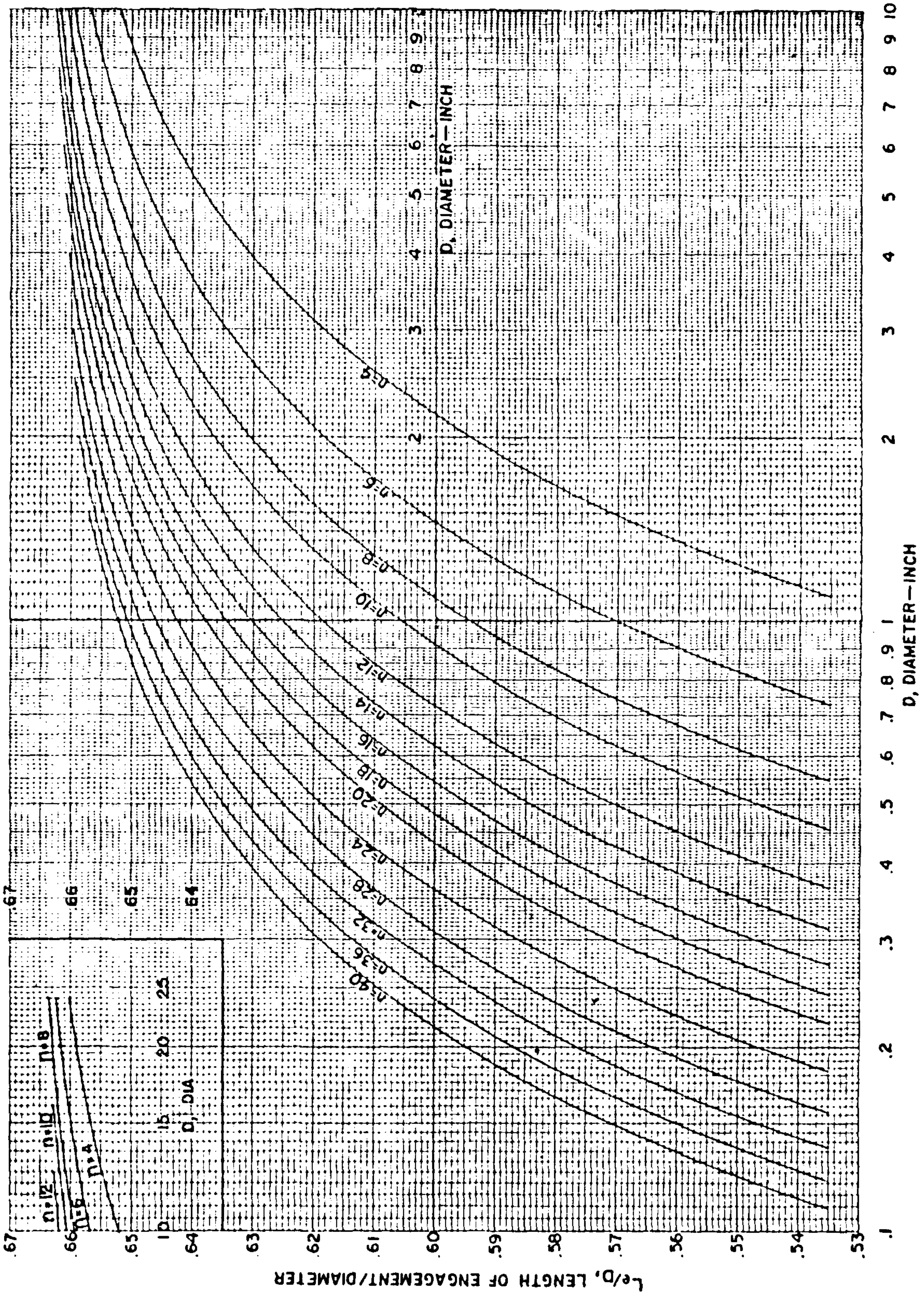
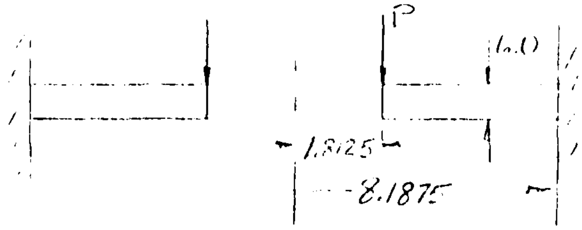


Figure A5.3. Chart for determining minimum length of thread engagement.

REF. NBS HANDBOOK H-28



$$P = \frac{\pi(1.8125)^2(60{,}000)}{2\,\pi(1.8125)} = 54{,}375 \text{ lbs/in.}$$

From "Formulas for Stress and Strain," R. J. Roark, 5th Ed., Table 24, case 1e, the maximum moment in the plate at the wall is:

$$M = -\,Pa\left(L_9 - \frac{C_7 L_6}{C_4}\right)$$

where: $$L_9 = \frac{r_o}{a}\left\{\frac{1+u}{2}\,\ell u\,\frac{a}{r_o} + \frac{1-u}{4}\left[1 - \left(\frac{r_o}{a}\right)^2\right]\right\}$$

$$L_6 = \frac{r_o}{4a}\left[\left(\frac{r_o}{a}\right)^2 - 1 + 2\,\ell u\,\frac{a}{r_o}\right]$$

$$C_4 = \frac{1}{2}\left[(1+u)\,\frac{b}{a} + (1-u)\,\frac{a}{b}\right]$$

$$C_7 = \frac{1}{2}\,(1-u^2)\left(\frac{a}{b} - \frac{b}{a}\right)$$

$r_o = 1.8125;$ $a = 8.1875;$ $b = 1.8125;$ $u = .3$

$\therefore\ L_9 = 0.254;\quad L_6 = 0.114;\quad C_4 = 1.725;\quad C_7 = 1.955$

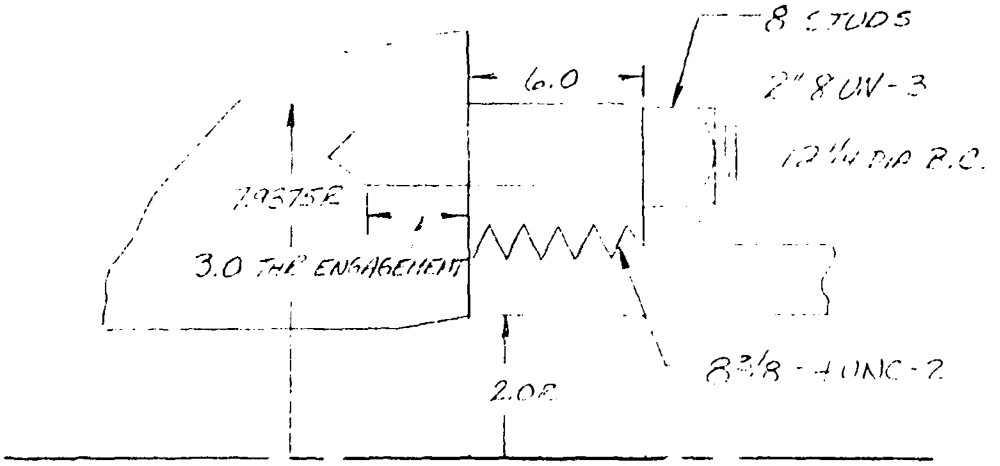
$$M = -(54{,}375)(8.1875)\left[0.254 - \frac{(1.955)(0.114)}{1.725}\right]$$

$M = 55{,}350$ in·lb/in.

$$\sigma_{MAX} = \frac{6M}{t^2} = 9{,}300 \text{ psi}$$

EXIT MANIFOLD FLANGE AND STUDS

A typical configuration of the Exit Manifold flange connection is shown below:



Following the procedure outlined in Section pressure load on studs and flange:

$$F = \pi R^2 P = \pi(2)^2(60,000) = 7.540 \times 10^5 \text{ lbs}$$

The tensile and shear areas of the studs are:

$$A_s = 0.5(C_1 K_n \text{min} \times \frac{L_e}{D} \times D_s \text{max})$$

$$S_s = K_n \text{max}(C_1 - C_5 T_{KN})$$

$$A_s = 0.5[(2.356)(1.865)(.6305)(2) = 2.770 \text{ in.}^2$$

$$S_s = (1.8797)\ [2.356 - (14.51)(.0147)] = 4.028 \text{ in.}^2$$

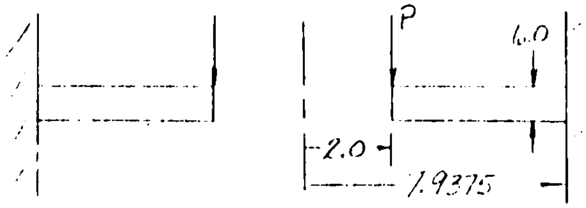
The tensile stress in the studs is:

$$\sigma = \frac{7.540 \times 10^5}{(8)(2.770)} = 34,000 \text{ psi}$$

The required thread engagement length is:

$$L_e = \frac{2A_s}{S_s} = \frac{(2)(2.77)}{4.028} = 1.375 < 3.0$$

Considering the flange as a circular plate, built-in at its edges:



$$P = \frac{\pi(2)^2(60,000)}{2\pi(2)} = 60,000 \text{ lbs/in.}$$

$$r_o = 2.0; \quad a = 7.9375; \quad b = 2.0; \quad u = 0.3$$

$$\therefore L_9 = 0.267; \quad L_6 = 0.115; \quad C_4 = 1.553; \quad C_7 = 1.691$$

$$M = -(60,000)(7.9375)\left[0.267 - \frac{(1.691)(0.115)}{1.553}\right]$$

$$M = 67,710 \text{ in}\cdot\text{lb/in.}$$

$$\sigma_{MAX} = \frac{6M}{t^2} = 11,300 \text{ psi}$$

APPENDIX 2A

PRIMARY STRESS EVALUATION
for
MACH 10 HEATER VESSEL

1. Primary Stresses in Cylinder Wall

Hand calculations were used to calculate the primary pressure stresses in the main vessel cylinder wall away from the threaded ends. These hand calculations are given on the following pages. The resulting stresses are listed and compared to the allowable stresses.

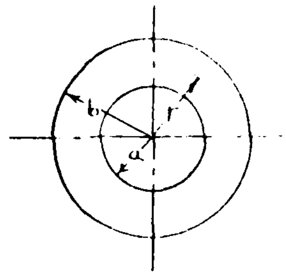
Primary Stresses in Cylinder

The P_m Stress Intensity is derived below:

$$\sigma_t = p\,\frac{a^2(b^2+r^2)}{r^2(b^2-a^2)} \quad \left\{\text{Tangential or Hoop stress}\right\}$$

$$\sigma_r = -p\,\frac{a^2(b^2-r^2)}{r^2(b^2-a^2)} \quad \left\{\text{Radial stress}\right\}$$

$$S = \sigma_t - \sigma_r \quad (\text{Stress Intensity})$$



The Average Stress Intensity is P_m:

$$P_m = \bar{\sigma}_t - \bar{\sigma}_r$$

$$\bar{\sigma}_t = \frac{1}{b-a}\int_{r=a}^{r=b} \sigma_t\,dr$$

$$\bar{\sigma}_r = \frac{1}{b-a}\int_{r=a}^{r=b} \sigma_r\,dr$$

$$P_m = \frac{1}{b-a}\int_{r=a}^{r=b} p\,\frac{a^2(b^2+r^2)}{r^2(b^2-a^2)}\,dr - \frac{1}{b-a}\int_{r=a}^{r=b} -p\,\frac{a^2(b^2-r^2)}{r^2(b^2-a^2)}\,dr$$

$$P_m = \frac{p\,a^2}{(b-a)(b^2-a^2)}\int_{r=a}^{r=b}\left(\frac{b^2+r^2}{r^2}+\frac{b^2-r^2}{r^2}\right)dr = \frac{2\,p\,a^2}{(b-a)(b^2-a^2)}\int_{r=a}^{r=b}\frac{b^2}{r^2}\,dr$$

$$\int_{r=a}^{r=b}\frac{b^2}{r^2}\,dr = \left[-\frac{b^2}{r}\right]_{r=a}^{r=b} = -\frac{b^2}{b}+\frac{b^2}{a} = b^2\,\frac{(b-a)}{ab} = \frac{b(b-a)}{a}$$

Therefore: $$P_m = \frac{2\,a\,b\,p}{(b^2-a^2)}$$

Primary Stresses in Cylinder (continued)

P_m Stress Intensity is:

$$P_m = \frac{2abp}{(b^2-a^2)} \quad \begin{cases} a = 14'' & b = 18.5'' \\ p = 15{,}000 \text{ psi} \end{cases}$$

$$P_m = \frac{2(14)(18.5)(15{,}000)}{\left[(18.5)^2-(14)^2\right]} = 53{,}128 \text{ psi}$$

The P_b Stress Intensity is derived below:

$$\sigma_t = p\,\frac{a^2(b^2+r^2)}{r^2(b^2-a^2)} \quad \text{(Tangential or Hoop Stress)}$$

$$\sigma_r = -p\,\frac{a^2(b^2-r^2)}{r^2(b^2-a^2)} \quad \text{(Radial Stress)}$$

$$S = \sigma_t - \sigma_r \quad \text{(Stress Intensity)}$$

For $a = 14''$, $b = 18.5''$ and $p = 15{,}000$ psi:

$$\sigma_t = \frac{15{,}000}{146.25}\left(\frac{14}{r}\right)^2(342.25 + r^2)$$

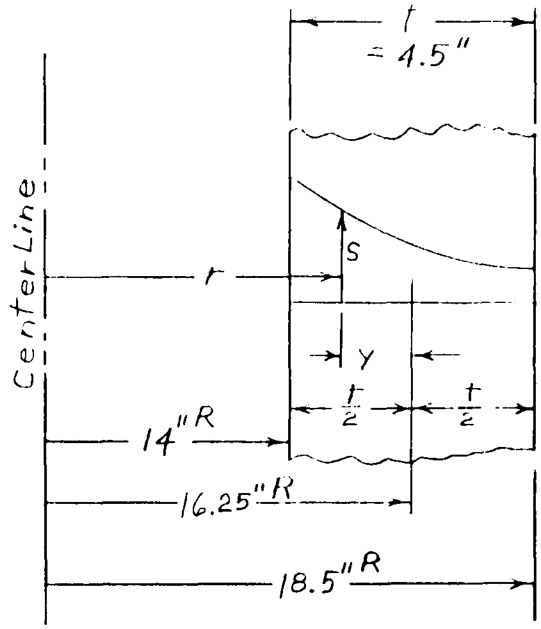
$$\sigma_r = \frac{-15{,}000}{146.25}\left(\frac{14}{r}\right)^2(342.25 - r^2)$$

$$S = \sigma_t - \sigma_r = \frac{15{,}000}{146.25}\left(\frac{14}{r}\right)^2(2)(342.25)$$

$$S = 70{,}205.12821\left(\frac{14}{r}\right)^2$$

Primary Stresses (continued)

$$P_b = \frac{6}{t^2}\int_{r=14}^{r=18.5} y S\,dr \qquad \left\{\sigma_b = \frac{6M}{t^2}\right)$$



$$y = 16.25 - r$$

$$S = 70,205.12821\left(\frac{14}{r}\right)^2$$

$$P_b = \frac{6}{t^2}\int_{14}^{18.5}(16.25 - r)(70,205.128)\left(\frac{14}{r}\right)^2$$

$$P_b = \frac{6(70,205.12821)(14)^2}{(4.5)^2}\int_{14}^{18.5}\left(16.25\frac{dr}{r^2} - \frac{dr}{r}\right)$$

$$P_b = 4,077,097.816\left[\frac{-16.25}{r} - \ln r\right]_{14}^{18.5}$$

$$P_b = 4,077,097.816\left[\frac{16.25}{14} - \frac{16.25}{18.5} - \ln\left(\frac{18.5}{14}\right)\right]$$

$$P_b = (4,077,097.816)(0.0036225048) = 14,769 \text{ psi}$$

Therefore, the Maximum $P_m + P_b$ Stress Intensity is:

$$P_m + P_b = 53,128 + 14,769 = 67,897 \text{ psi}$$

Primary Pressure Stresses in Cylinder Compared to the Allowable Stresses

Stress Category	Calculated stress, psi	Allowable Stress, psi
P_m	53,128	$S_m = 67,500$ psi
$P_m + P_b$	67,897	$1.5 S_m = 101,200$ psi

$S_u = 135,000$ psi

$$S_m = \frac{S_u}{2} = 67,500 \text{ psi}$$

Stresses in Cylinder are due to an internal Pressure of 15,000 psi.

APPENDIX 2B

FATIGUE EVALUATION OF THREADS
ON RIGHT END CLOSURE

of

MACH 10 HEATER VESSEL

FATIGUE EVALUATION OF THREADS

The fatigue analysis calculations used to calculate the fatigue design life of the threads on the right end closure of the MACH 10 Heater Vessel are given on the following pages. The calculations are divided into the following parts:

(a) Summary of Loads on Main Cylinder Threads

(b) Equivalent Pressure Calculation for Maximum Thread Load

(c) Edge Displacements for Detailed Model

(d) Maximum Stress Intensities and Maximum Displacements in Thread Subjected to Highest Thread Load

(e) Fatigue Analysis of Stress Gradient at Thread Root Radius

(f) Fatigue Life of Threads on Right End Closure

(g) Fatigue Curve for Body Material of the MACH 10 Heater Vessel

As shown below, a fatigue design life of 640 cycles was obtained for the threads on the right end closure.

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY E.E.P DATE 1-/4/77 SUBJECT MACH 10 Heater Vessel SHEET NO 1 OF 6
CHKD. BY DATE PROJ. NO TP/E70

Forces on Main Cylinder Threads - from Overall Model
Internal Pressure Stress Run

Tooth	Element	Node	F_x (Lbs/rad)		F_y (Lbs/rad)	
21	21	45	-0.962082 E+3	0	-0.165520 E+4	-0.306213 E+4
	22	45	0.962082 E+3		-0.140693 E+4	
	22	46	0.155087 E+4	0	0.259832 E+4	0.801381 E+4
	23	46	-0.155087 E+4		0.541549 E+4	
	23	47	0.198608 E+4	0	0.357984 E+4	0.1284991 E+5
	24	47	-0.198608 E+4		0.927007 E+4	
20	48	81	-0.596861 E+3	0	-0.803403 E+3	-0.7578509 E+3
	49	81	0.596861 E+3		0.455521 E+2	
	49	82	0.145508 E+4	0	0.327639 E+4	0.933403 E+4
	50	82	-0.145508 E+4		0.605764 E+4	
	50	83	0.147083 E+4	0	0.360960 E+4	0.1328973 E+5
	51	83	-0.147083 E+4		0.968013 E+4	
19	75	117	-0.136442 E+3	0	0.335759 E+3	0.2473319 E+4
	76	117	0.136442 E+3		0.213756 E+4	
	76	118	0.141846 E+4	0	0.454351 E+4	0.1207156 E+5
	77	118	-0.141846 E+4		0.752805 E+4	
	77	119	0.857928 E+3	0	0.399349 E+4	0.1504429 E+5
	78	119	-0.857928 E+3		0.110508 E+5	
18	103	154	-0.152347 E+4	0	-0.277461 E+4	-0.142245 E+4
	104	154	0.152347 E+4		0.135216 E+4	
	104	155	0.645422 E+4		0.948781 E+4	0.3469101 E+5
	105	155	-0.645422 E+4		0.252032 E+5	
17	129	188	0.745058 E-7		0.156454 E+4	
	129	189	0.249836 E+3	0	0.287740 E+4	0.877773 E+4
	130	189	-0.249836 E+3		0.590033 E+4	
	130	190	0.620756 E+3	0	0.657870 E+4	0.1647914 E+5
	131	190	-0.620756 E+3		0.990044 E+4	
	131	191	-0.121020 E+4	0	0.433175 E+4	0.1711735 E+5
	132	191	0.121020 E+4		0.127856 E+5	
16	156	224	0.447035 E-7		0.282274 E+4	
	156	225	0.855257 E+2	0	0.377907 E+4	0.1073826 E+5
	157	225	-0.855257 E+2		0.695919 E+4	
	157	226	0.246827 E+3	0	0.739762 E+4	0.1840782 E+5
	158	226	-0.246827 E+3		0.110102 E+5	
	158	227	-0.199110 E+4	0	0.462728 E+4	0.1848698 E+5
	159	227	0.199110 E+4		0.138597 E+5	

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY PBP DATE 12/9/77 SUBJECT MACH 10 Heater Vessel SHEET NO 2 OF 6

CHKD. BY DATE PROJ. NO JP1270

Forces on Main Cylinder (continued)

Tooth	Element	Node	Fx (Lbs/rad)		Fy (Lbs/rad)	
15	183	260	0.447035E-7		0.450992E+4	
	183	261	-0.188451E+3	0	0.484394E+4	0.1292791E+5
	184	261	0.188451E+3		0.808397E+4	
	184	262	-0.273821E+3	0	0.826102E+4	0.2045502E+5
	185	262	0.273821E+3		0.121940E+5	
	185	263	-0.300400E+4	0	0.489416E+4	0.1991696E+5
	186	263	0.300400E+4		0.150228E+5	
14	210	296	0.447035E-7		0.656427E+4	
	210	297	-0.555010E+3	0	0.604029E+4	0.1525994E+5
	211	297	0.555010E+3		0.921965E+4	
	211	298	-0.929461E+3	0	0.708937E+4	0.2239977E+5
	212	298	0.929461E+3		0.133104E+5	
	212	299	-0.419633E+4	0	0.508213E+4	0.2115063E+5
	213	299	0.419633E+4		0.160685E+5	
13	237	332	0.596046E-7		0.880692E+4	
	237	333	-0.959046E+3	0	0.731687E+4	0.1768877E+5
	238	333	0.959046E+3		0.103719E+5	
	238	334	-0.163107E+4	0	0.991543E+4	0.2431203E+5
	239	334	0.163107E+4		0.143966E+5	
	239	335	-0.548071E+4	0	0.521265E+4	0.2238145E+5
	240	335	0.548071E+4		0.171688E+5	
12	264	368	0.447035E-7		0.115550E+5	
	264	369	-0.149449E+4	0	0.875315E+4	0.2022665E+5
	265	369	0.149449E+4		0.114735E+5	
	265	370	-0.253706E+4	0	0.106036E+5	0.258513E+5
	266	370	0.253706E+4		0.152477E+5	
	266	371	-0.701962E+4	0	0.515986E+4	0.2308776E+5
	267	371	0.701962E+4		0.179279E+5	
11	291	404	0.596046E-7		0.146242E+5	
	291	405	-0.208990E+4	0	0.103399E+5	0.229687E+5
	292	405	0.208990E+4		0.126288E+5	
	292	406	-0.354058E+4	0	0.112855E+5	0.273285E+5
	293	406	0.354058E+4		0.160430E+5	
	293	407	-0.873706E+4	0	0.500487E+4	0.2370837E+5
	294	407	0.873706E+4		0.187035E+5	
10	318	440	0.447035E-7		0.185052E+5	
	318	441	-0.288955E+4	0	0.122048E+5	0.259552E+5
	319	441	0.288955E+4		0.137504E+5	
	319	442	-0.488410E+4	0	0.117738E+5	0.282751E+5
	320	442	0.488410E+4		0.165013E+5	
	320	443	-0.108986E+5	0	0.455973E+4	0.2351483E+5
	321	443	0.108986E+5		0.189551E+5	

ENGINEERING DESIGN & ANALYSIS SERVICES

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PITTSBURGH, PENNSYLVANIA

BY DEP DATE 1-19/77 SUBJECT MACH 10 Heater Vessel SHEET NO 3 OF 6
CHKD. BY DATE PROJ. NO JP1270

Forces on Main Cylinder (continued)

Tooth	Element	Node	F_X (Lbs/rad)		F_Y (Lbs/rad)	
9	345	476	0.298023E-7		0.229408E+5	
	345	477	-0.377468E+4	0	0.143962E+5	0.295206E+5
	346	477	0.377468E+4		0.151244E+5	
	346	478	-0.639376E+4	0	0.123968E+5	0.294757E+5
	347	478	0.639376E+4		0.170789E+5	
	347	479	-0.133908E+5	0	0.404373E+4	0.2345033E+5
	348	479	0.133908E+5		0.194064E+5	
8	372	512	0.149012E-7		0.286700E+5	
	372	513	-0.496697E+4	0	0.170915E+5	0.336882E+5
	373	513	0.496697E+4		0.165967E+5	
	373	514	-0.840966E+4	0	0.128888E+5	0.302700E+5
	374	514	0.840966E+4		0.173812E+5	
	374	515	-0.166040E+5		0.317859E+4	0.2258049E+5
	375	515	0.166040E+5		0.194019E+5	
7	399	548	0.149012E-7		0.353414E+5	
	399	549	-0.633744E+4	0	0.202605E+5	0.386014E+5
	400	549	0.633744E+4		0.183409E+5	
	400	550	-0.106919E+5	0	0.135182E+5	0.312839E+5
	401	550	0.106919E+5		0.177657E+5	
	401	551	-0.202689E+5	0	0.219970E+4	0.218243E+5
	402	551	0.202689E+5		0.196246E+5	
6	426	584	0.149012E-7		0.439423E+5	
	426	585	-0.838830E+4	0	0.235175E+5	0.422521E+5
	427	585	0.838830E+4		0.187346E+5	
	427	586	-0.139262E+5	0	0.125014E+5	0.281636E+5
	428	586	0.139262E+5		0.156622E+5	
	428	587	-0.247817E+5	0	-0.165880E+3	0.1658562E+5
	429	587	0.247817E+5		0.167515E+5	
5	453	620	0.149012E-7		0.530463E+5	
	453	621	-0.110544E+5	0	0.254203E+5	0.409403E+5
	454	621	0.110544E+5		0.155220E+5	
	454	622	-0.180835E+5		0.752094E+4	0.1512425E+5
	455	622	0.180835E+5		0.760331E+4	
4	480	656	0.745058E-8		0.601459E+5	
	480	657	-0.159650E+5	0	0.205348E+5	0.230720E+5
	481	657	0.159650E+5		0.253720E+4	
3	507	692	-0.745058E-8		0.655721E+5	
	507	693	-0.196679E+5	0	0.171583E+5	0.1213643E+5
	508	693	0.196679E+5		-0.502187E+4	
2	534	728	-0.745058E-8		0.731292E+5	
	534	729	-0.228488E+5	0	0.169790E+5	0.810586E+4
	535	729	0.228488E+5		-0.887314E+4	

ENGINEERING DESIGN & ANALYSIS SERVICES — O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY LFi DATE 12/1/77 SUBJECT MACH 10 Heater Vessel SHEET NO 4 OF 6
CHKD. BY DATE PROJ. NO JF127C

Forces on Main CyLinder (continued)

Tooth	ELement	Node	Fx (Lbs/rad)		Fy (Lbs/rad)	
1	561	764	0		-0.745058E-7	
	561	765	0.115518E+4	0	0.749501E+3	0
	562	765	-0.115518E+4		-0.749501E+3	
	562	766	0.469114 E+4	0	0.168592E+4	0
	563	766	-0.469114E+4		-0.168592E+4	
	563	767	0.108065E+4	0	-0.453386E+4	0
	564	767	-0.108065E+4		0.453386E+4	

Check By Examining Some Plug Threads

Thread	ELement	Node	Fx (Lbs/rad)		Fy (Lbs/rad)	
12	979	1375	-0.135637E+5	0	-0.785809E+4	-0.8806951E+4
	980	1375	0.135637E+5		-0.948831E+3	
	980	1376	-0.798896E+4	0	-0.101323E+5	-0.1768879E+5
	981	1376	0.798896E+4		-0.755649E+4	
	981	1377	-0.421097E+4	0	-0.115193E+5	-0.243121E+5
	982	1377	0.421097E+4		-0.127928E+5	
	982	1378	-0.461936E-6		-0.223814E+5	
5	1214	1655	-0.911764E+4	0	-0.353340E+5	-0.4394236E+5
	1215	1655	0.911764E+4		-0.860836E+4	
	1215	1656	-0.188707E+4	0	-0.251302E+5	-0.422521E+5
	1216	1656	0.188707E+4		-0.171219E+5	
	1216	1657	-0.230760E+4	0	-0.159466E+5	-0.281636E+5
	1217	1657	0.230760E+4		-0.122170E+5	
	1217	1658	-0.201166E-6		-0.165856E+5	

Summary of Forces on Main Cylinder Threads

Thread	$\sum F_y$ (Lbs/rad) $\times 10^5$
21	0.1780159
20	0.218659091
19	0.29589169
18	0.3326856
17	0.4393876
16	0.504558
15	0.5780981
14	0.6537461
13	0.7318917
12	0.8072071
11	0.8862977
10	0.9625033
9	1.0538743
8	1.1520869
7	1.27051
6	1.3094362
5	1.0911085
4	0.832179
3	0.7770853
2	0.8123506
1	0

← Max, (No. 6)

$$\sum F_y(\text{Total}) = 14.88757268 \times 10^5 \text{ Lbs/rad}$$

$$\left[\sum F_y(\text{Total})\right] \cdot \cos(5.03^\circ) = 14.83 \times 10^5 \text{ Lbs/rad}$$

$p = 15{,}000$ psi

$$F_p = \frac{(15{,}000)\cancel{\pi}(28.125)^2}{4(2\cancel{\pi})} = 14.83 \times 10^5 \text{ lbs/rad}$$

Agree!

$$\sum F_y(\text{Ave}) = \frac{\sum F_y(\text{Total})}{20} = 0.744378634 \times 10^5 \text{ Lbs/rad}$$

$$\frac{\sum F_y(\text{Max})}{\sum F_y(\text{Ave})} = 1.7591$$

ENGINEERING DESIGN & ANALYSIS SERVICES

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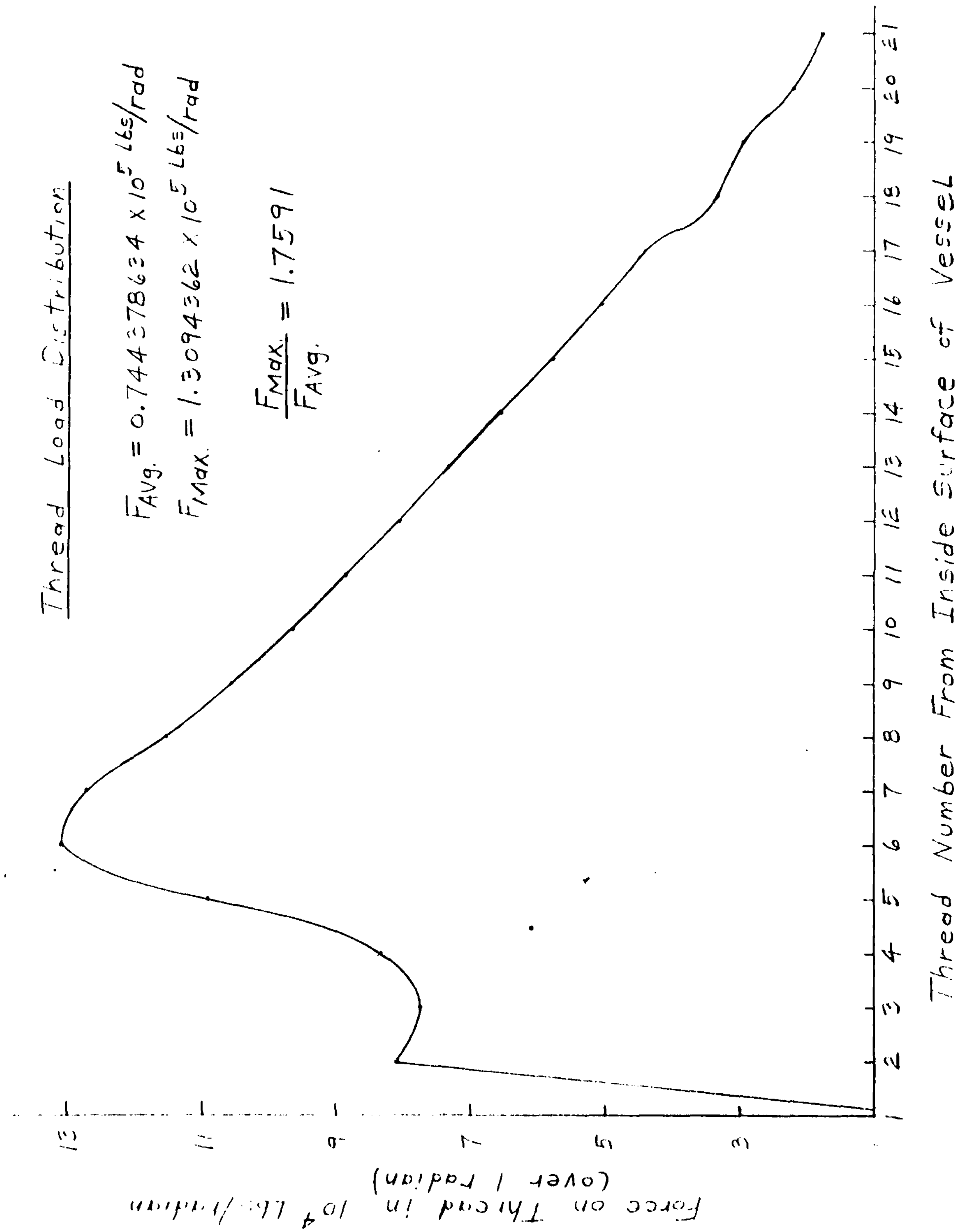
BY DBP DATE 12/9/77 SUBJECT MACH 10 Heater Vessel SHEET NO 6 OF 6

CHKD. BY DATE PROJ. NO JP1270



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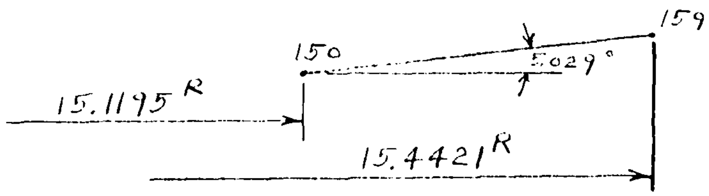
BY DEP DATE 1-/4/77 SUBJECT MACH 10 Heater Vessel SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JF1270

Total Force on Tooth # 6 (body) from the overall Model = 1.3094362×10^5 lbs/rad

Total $F = 2\pi(1.3094362 \times 10^5)$ lbs



$$\text{Area} = \frac{\pi\left[(15.4421)^2 - (15.1195)^2\right]}{\cos(5.029°)}$$

$$\text{Max. Pressure} = \frac{F}{\text{Area}} = \frac{2\cancel{\pi}(1.3094362 \times 10^5)\cdot\cos(5.029°)}{\cancel{\pi}\left[(15.4421)^2 - (15.1195)^2\right]}$$

$$= 26{,}460.548 \text{ psi}$$

$\therefore$ P = 26,461 psi on Face 1 of Elements 150, 151, 152, 153, 154, 155, 156, 157, 158.

ENGINEERING DESIGN & ANALYSIS SERVICES

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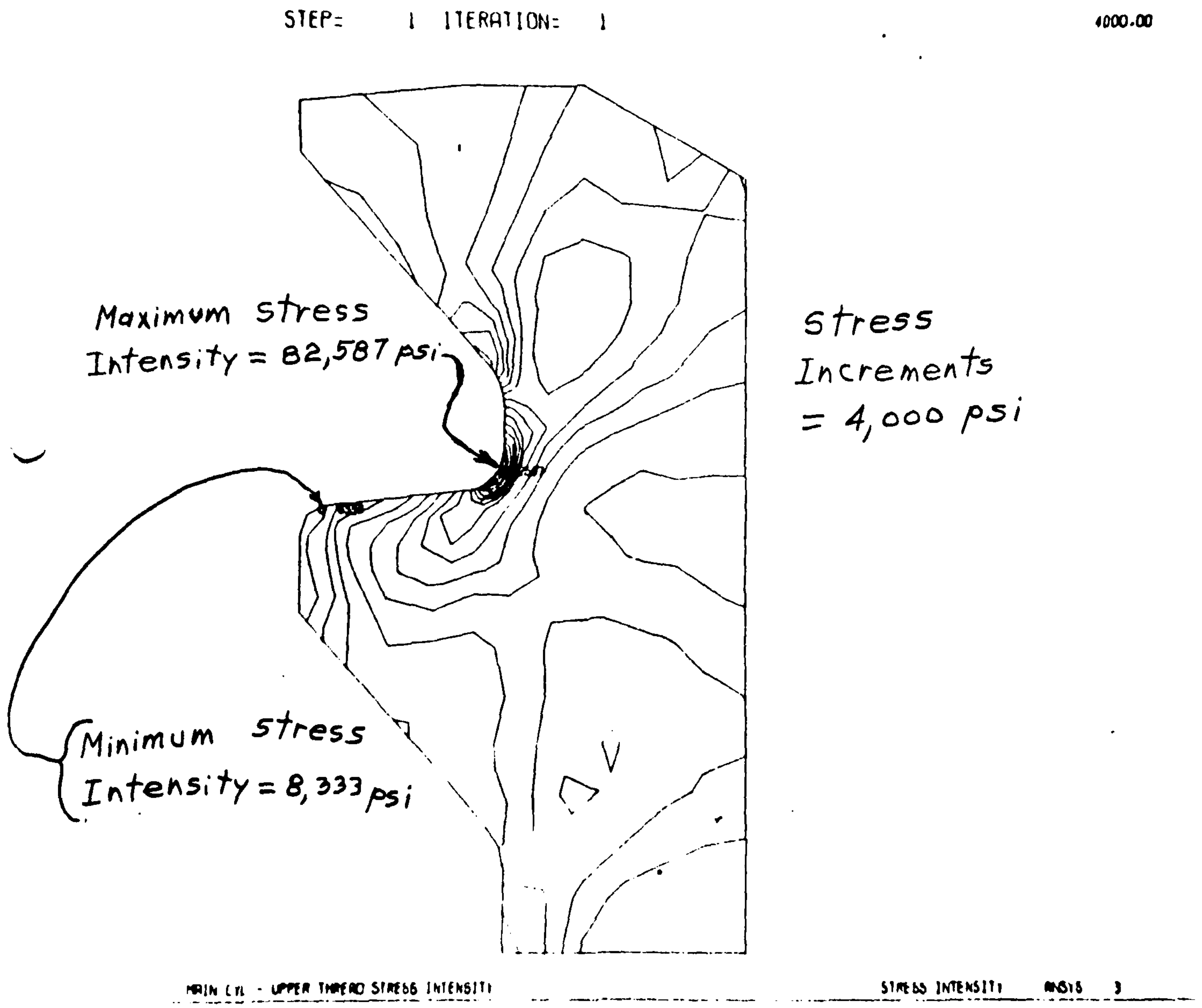
BY DBP DATE 12/8/77 SUBJECT MACH 10 Heater Vessel SHEET NO 1 OF 1

CHKD. BY DATE PROJ. NO JH270

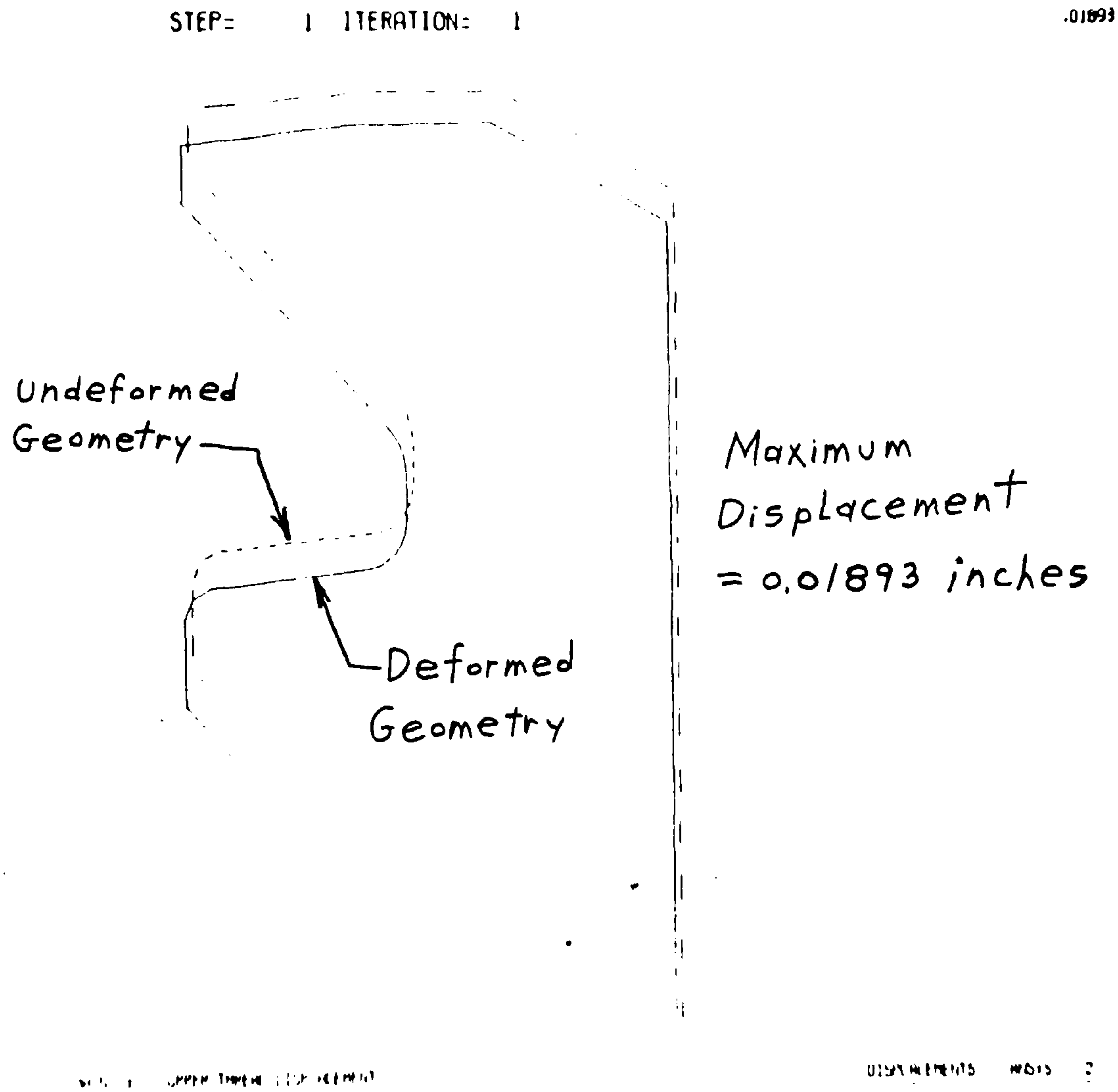
Detail Model Edge Displacements

Node		Coordinates		Displacements	
Overall Model	Detail Model	X (in)	Y (in)	δ_x (in)	δ_y (in)
*552	1	15.5625	1.0	-0.00347734	-0.0183424
	2	15.6625	1.0	-0.00345275	-0.0182092
*553	3	15.7625	1.0	-0.00342816	-0.0180760
	4	15.8625	1.0	-0.003425885	-0.0179951
	5	15.9625	1.0	-0.00342361	-0.0179142
	6	16.0625	1.0	-0.003421335	-0.0178333
*554	7	16.1625	1.0	-0.00341906	-0.0177524
	14		1.113	-0.00347486	-0.0176792
*559	21		1.23	-0.00353264	-0.0176025
	28		1.34	-0.00359006	-0.0175153
	38		1.443	-0.00364382	-0.0174327
	48		1.546	-0.00369759	-0.0173500
*571	58		1.673	-0.00376380	-0.0172481
	70		1.7536	-0.00380715	-0.0171887
	93		1.85	-0.003858697	-0.0171177
	99		1.952	-0.00391365	-0.0170426
*590	105		2.0	-0.00393942	-0.0170072
	111		2.088	-0.00397093	-0.01693553
	117		2.129	-0.00398561	-0.01690213
	123		2.17622	-0.00400252	-0.01686367
*595	129		2.228	-0.00402106	-0.0168215
	135		2.335	-0.00405942	-0.0167197
	141		2.4	-0.00408273	-0.01665781
	147		2.5	-0.00411858	-0.01656264
	303		2.6	-0.00415444	-0.01646747
*607	324	16.1625	2.673	-0.00418061	-0.0163980
	323	16.1102	2.6995	-0.00408496	-0.01639776
	322	16.0032	2.7539	-0.00419389	-0.01639728
	321	15.8888	2.812	-0.00420342	-0.01639677
*619	320	15.7625	2.8761	-0.00421394	-0.0163962
	319	15.6625	2.8761	-0.004210735	-0.0165284
*618	318	15.5625	2.8761	-0.00420753	-0.0166606
*617	317	15.45	2.8761	-0.00421450	-0.0169112
*616	316	15.3525	2.86752	-0.00425099	-0.0172807
*615	315	15.2075	2.85476	-0.00429021	-0.0177706
*614	314	15.0625	2.842	-0.004[illegible]288	-0.0182694

* Coordinates and displacements at these nodes come from run ΦDANDKU-12/8/77. All other displacements are linearly interpolated.



Computer Drawn Stress Contour Plot



Computer Drawn Deformation Plot

ENGINEERING DESIGN & ANALYSIS SERVICES

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PITTSBURGH, PENNSYLVANIA

BY DBP DATE 12/13/77 SUBJECT MACH 10 Heater Vessel SHEET NO 1 OF 8

CHKD. BY DATE PROJ. NO JP1270

Determine Material Constant, δ

The stress distribution across a section containing a cir-

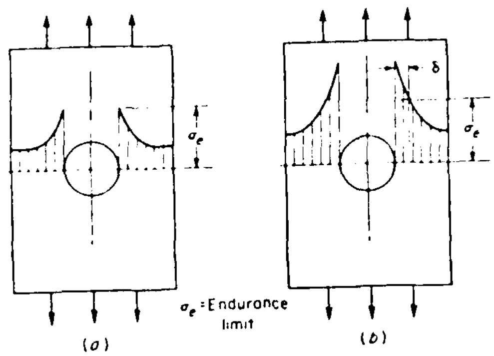

Fig. 6.29. Stress Distribution about a Circular Hole in a Bar

cular hole, Fig. 6.29*a*, has a high stress gradient at the edge of the hole. If the load is just sufficient to bring the peak stress up to the endurance limit, a fatigue failure would hardly be expected since the volume of material at this stress is zero. A finite volume of material must be at the endurance limit before a crack will form, and to obtain this volume of material the endurance limit stress must exist at some finite depth, δ_1, below the surface; therefore, the steeper the stress gradient, the higher the load required to produce fatigue failure, Fig. 6.29*b*.

The dimension, δ, is a property of the material; and, in general, hard, fine-grained materials have small values of δ, whereas soft, coarse-grained materials have larger values. The relationship between δ and steel tensile strengths, based on correlating fatigue data and the shear theory of failure is shown in Fig. 6.30.

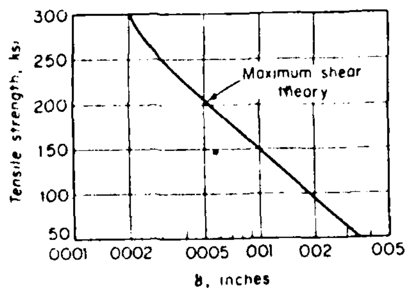

Fig. 6.30. Material Constant δ vs. Tensile Strength for Steel

For the body material, the Tensile strength is 135 Ksi and δ is Equal to 0.00125 inches.

Calculate Stress Intensity at Depth δ

Ref: Timoshenko and Goodier, Theory of Elasticity, p. 90

The stress distribution in the vicinity of a small circular hole in the middle of a plate subjected to uniform Tension is given by:

$$\sigma_r = S/2\left[1-(a/r)^2\right] + S/2\left[1+3(a/r)^4-4(a/r)^2\right]\cos 2\theta$$

$$\sigma_\theta = S/2\left[1+(a/r)^2\right] - S/2\left[1+3(a/r)^4\right]\cos 2\theta$$

$$\tau_{r\theta} = -S/2\left[1-3(a/r)^4+2(a/r)^2\right]\sin 2\theta$$

When $\theta=0$, $\tau_{r\theta}=0$ and the principal stresses are:

$$\sigma_r = S/2\left[2+3(a/r)^4-5(a/r)^2\right]$$

$$\sigma_\theta = S/2\left[-3(a/r)^4+(a/r)^2\right]$$

The stress Intensity is given by:

$$S.I. = |\sigma_r-\sigma_\theta| = S/2\left[2+6(a/r)^4-6(a/r)^2\right]$$

$$= S\left[1+3(a/r)^4-3(a/r)^2\right]$$

Assume that the stress intensity distribution at the thread root radius has the same form as the above stress intensity distribution:

$$S.I. = S\left[1+A(a/r)^4-B(a/r)^2\right]$$

Where: a = Thread Root Radius = 0.09375 in.

$r = a + \delta$, in.

δ = Distance from surface, in.

S, A and B are three unknown constants.

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
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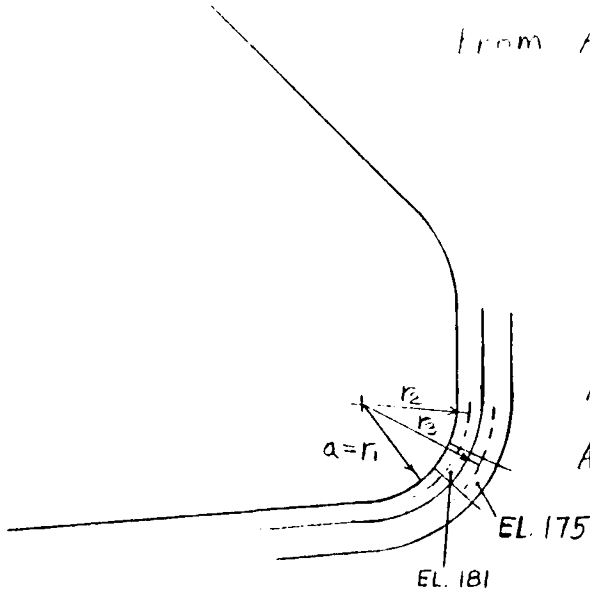
BY DEF DATE 12/13/77 SUBJECT Mach to Heater Vessel SHEET NO 5 OF 8
CHKD BY DATE PROJ. NO JT1270



From ANSYS Run ΦDANDTT, 12/12/77

$a = 0.09375 \text{ in} = r_1$

$r_2 = 0.105375 \text{ in.}$

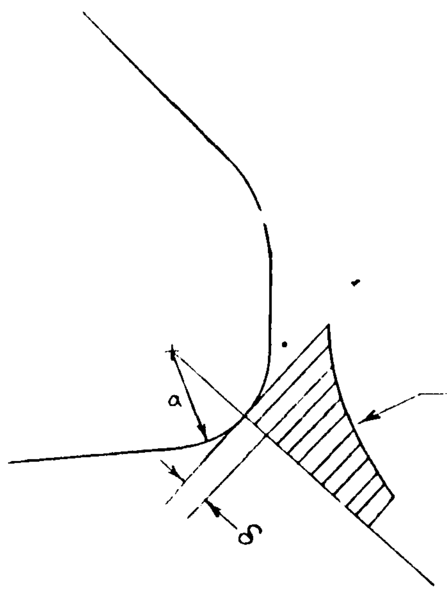
$r_3 = 0.1312 \text{ in.}$

At $(a/r_1) = 1$, S.I. = 114,969 psi

At $(a/r_2) = 0.88968$, S.I. = 93,128 psi

At $(a/r_3) = 0.71456$, S.I. = 62,697 psi

The Known Stress Intensities at the above three Locations can be used to evaluate the three unKnowns in the Stress Intensity Distribution Equation.



$$S.I. = S\left[1 + A(a/r)^4 - B(a/r)^2\right]$$

(1) $(a/r) = 1$ $\quad 114{,}969 = S(1 + A - B)$

(2) $(a/r) = 0.88968$ $\quad 93{,}128 = S(1 + 0.62652A - 0.79153B)$

(3) $(a/r) = 0.71456$ $\quad 62{,}697 = S(1 + 0.26071A - 0.51060B)$

From (1): $\quad S = \dfrac{114{,}969}{1 + A - B}$

(1) into (2):

$$93{,}128 = \frac{114{,}969}{1 + A - B}\left(1 + 0.62652A - 0.79153B\right)$$

$$93{,}128 + 93{,}128A - 93{,}128B = 114{,}969 + 72{,}030.3779A - 91{,}001.4126B$$

$$21{,}097.6221A - 2{,}126.5874B = 21{,}841$$

$$9.920881738A - B = 10.2704455$$

(1) into (3):

$$62{,}697 + 62{,}697A - 62{,}697B = 114{,}969 + 29{,}973.568A - 58{,}703.1714B$$

$$32{,}723.432A - 3{,}993.8286B = 52{,}272$$

$$8.19349934A - B = 13.08819312$$

$$\begin{array}{rcl} 9.920881738A - B &=& 10.2704455 \\ -(8.19349934A - B &=& 13.08819312) \\ \hline 1.727382398A &=& -2.81774762 \\ A &=& -1.631223997 \\ B &=& -26.45362586 \\ S &=& 4{,}452.296909 \end{array}$$

$$S.I. = 4{,}452.296909\left[1 + 26.45362586(a/r)^2 - 1.631223997(a/r)^4\right]$$

At $(a/r) = 1$, $S.I. = 114{,}969$ psi

At $(a/r) = 0.88968$, $S.I. = 93{,}128$ psi

At $(a/r) = 0.71456$, $S.I. = 62{,}697$ psi

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CHKD BY DATE PROJ. NO JP1270

At $r = a + 0.00125 = 0.09375 + 0.00125 = 0.095$ in.

$$\frac{a}{r} = \frac{0.09375}{0.095} = 0.9868$$

And $S.I. = 4,452.296909\left[1 + 26.4536(0.9868)^2 - 1.6312(0.9868)^4\right]$

$= 112,265$ psi

Therefore, the stress Intensity at the root of Thread No. 6 on the body where the thread Load is a maximum and equal to 1.3094362×10^5 lbs/rad is:

$S.I._{(Max)} = 112,265$ psi

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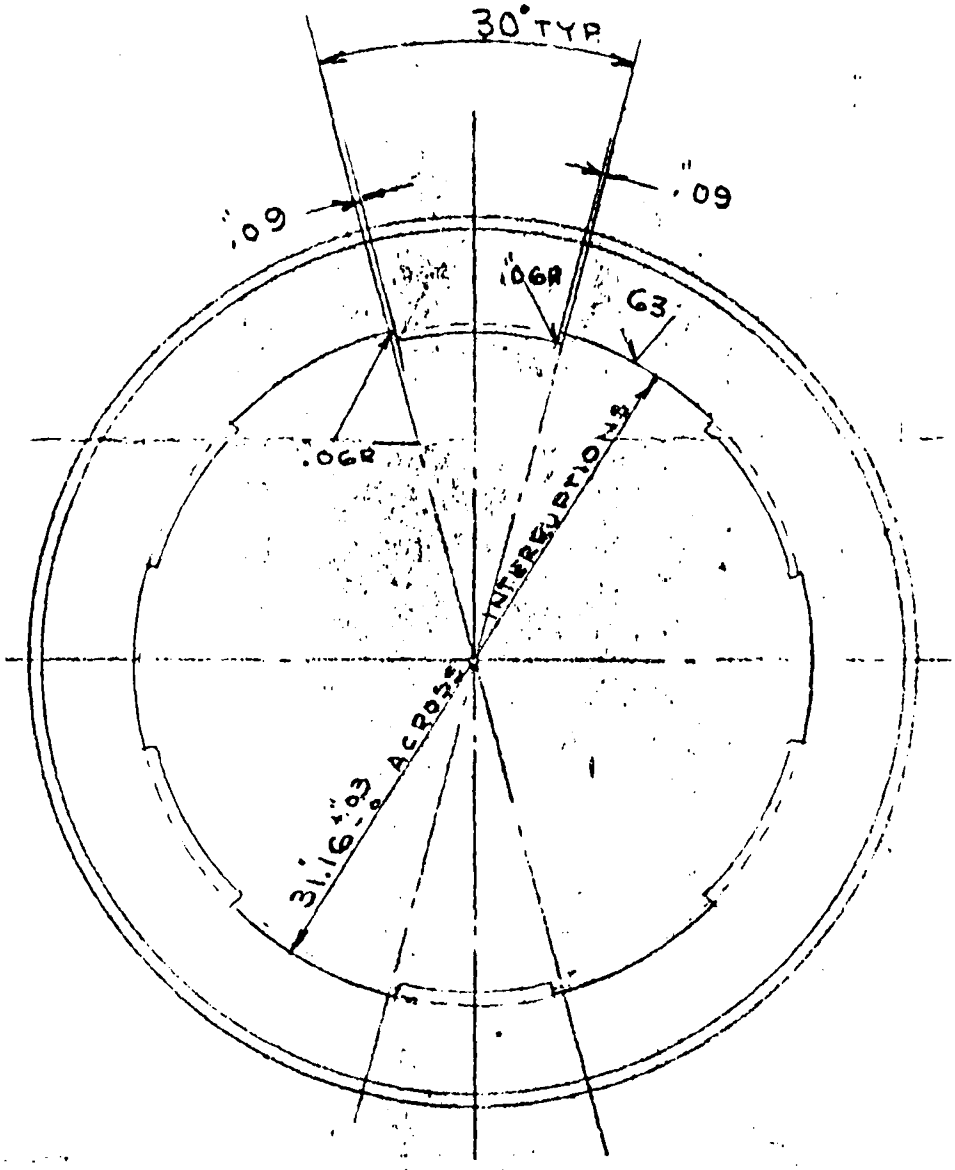
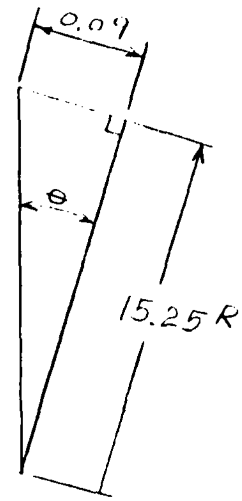
CHKD. BY DATE PROJ. NO JP1570

Right End Closure has Interrupted Threads:



Computer Results are for Continuous Threads. Therefore, Force and stress on these interrupted threads will be calculated by rationing up the results for continuous threads.

Equivalent Force on Interrupted Thread



$$\theta = Tan^{-1}\left(\frac{0.09}{15.25}\right) = 0.338135°$$

$$2\theta = 0.67627°$$

$$F_{eq} = F\left(\frac{60}{30 - 0.67627}\right)$$

$$F_{eq} = 2.04612\ F$$

Therefore, the Maximum stress in the interrupted threads is:

$$\sigma_{Max} = 2.04612(112,265) = 229,708 \text{ psi}$$

Fatigue Life of Threads on Right End Closure

$S_{range}\,(Max) = 229{,}708$ psi

$S_{alt} = 114{,}854$ psi $\qquad S_y = 120{,}000$ psi

$S'_{mean} = 114{,}854$ psi $\qquad S_u = 135{,}000$ psi

$S_{alt} + S'_{mean} = 229{,}708$ psi

$S_{alt} < S_y$ and $S_{alt} + S'_{mean} > S_y$

$\therefore\ S_{mean} = S_y - S_{alt}$

$S_{mean} = 120{,}000 - 114{,}854 = 5{,}146$ psi

$$S_{eq} = \frac{7(114{,}854)}{8 - \left[1 + \frac{5{,}146}{135{,}000}\right]^3} = 116{,}836 \text{ psi}$$

The Design Life from ASME Paper No. 76-PVP-62 for ASTM A-723, CL. 2 Material with a Factor of 2 on Stress and a factor of 20 on Cycles is:

$N = 640$ Cycles $\boxed{\text{Design Life}}$

Since the theoretical fatigue curves from this paper were obtained on small polished specimens tested in air, factors must be applied to account for size effects, surface finish, environmental effects, and scatter of data. Therefore, a factor of either 2 on stress or 20 on cycles, whichever is more conservative at each point, was applied to the mean failure curve to obtain a design curve which accounts for these effects. These factors have been confirmed by several fatigue tests and simulated service tests on models of components.

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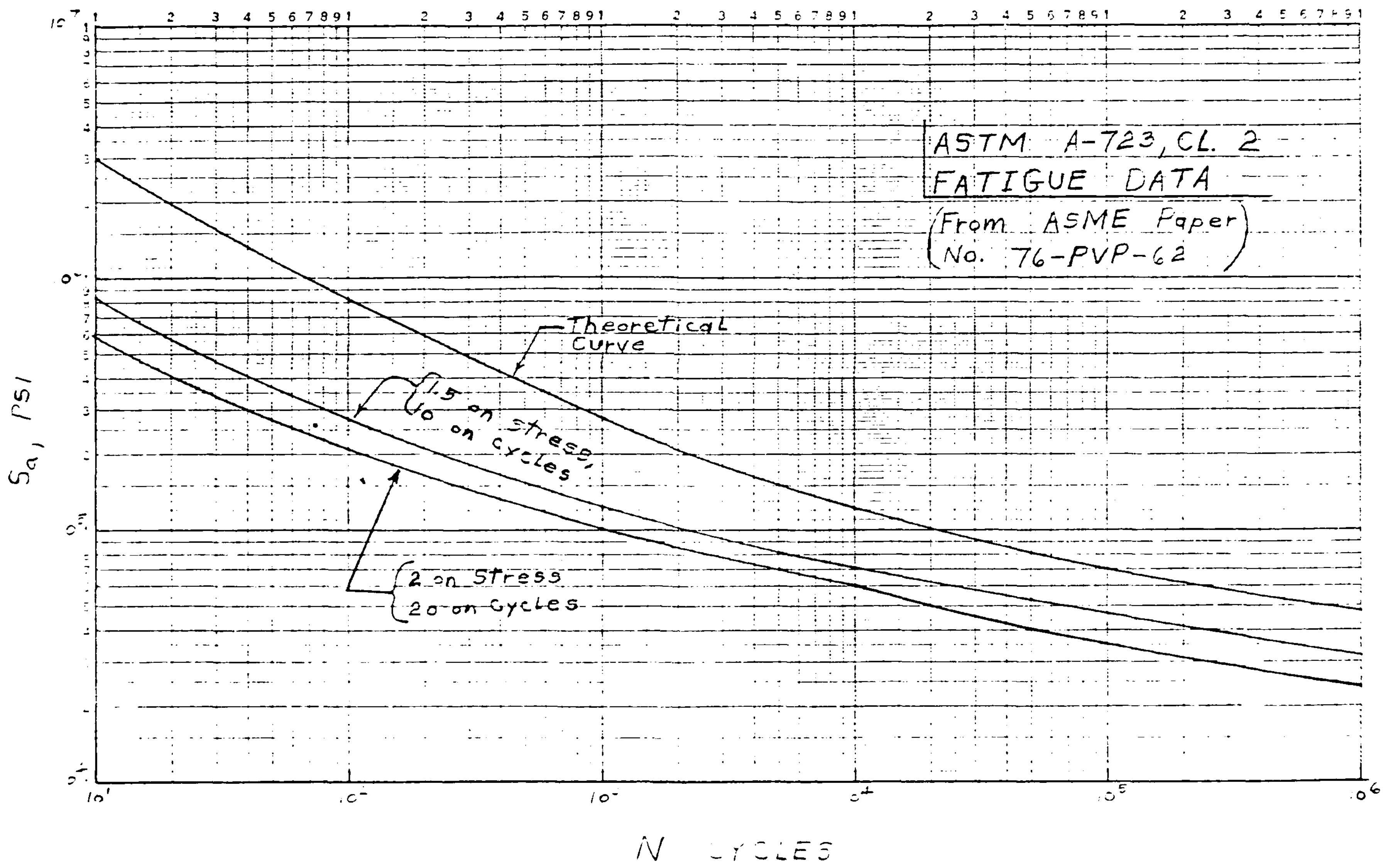

APPENDIX 2C

FRACTURE MECHANICS EVALUATION OF THREADS ON
RIGHT END CLOSURE OF MACH 10 HEATER VESSEL

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BY DBP DATE 12/19/77 SUBJECT MACH 10 Heater Vessel SHEET NO 1 OF 5
CHKD. BY DATE PROJ. NO JP1270

Crack Growth Rate Analysis of Threads on the MACH 10 Heater Vessel:

REFERENCES:

(1) Imhof, E. J. and Barsom, J. M., "Fatigue and Corrosion-Fatigue Crack Growth of 4340 Steel At Various Yield Strengths", Progress in Flaw Growth and Fracture Toughness Testing, ASTM STP 536, American Society for Testing and Materials, 1973, pp. 182-205.

(2) Wessel, E. T. and Mager, T. R., "Fracture Mechanics Technology As Applied to Thick-Walled Nuclear Pressure Vessels", Proc. Conf. on Practical Application of Fracture Mechanics to Pressure Vessel Technology, Institution of Mechanical Engineers, 1971.

BASIC ASSUMPTIONS

1. Thread Material is modified AISI 4340, or "gun steel". This is now designated ASTM A-723, Class 2 Material. Assume this Material has the following Properties:

$$S_U = 135{,}000 \text{ psi}$$

$$S_Y = 120{,}000 \text{ psi}$$

$$K_{IC} = 100 \text{ Ksi}\sqrt{\text{in}}$$

2. From Reference (1), the Crack growth rate for this material is represented by the following Equation:

$$\frac{da}{dN} = 0.66 \times 10^{-8} (\Delta K)^{2.25}$$

Where: $\frac{da}{dN}$ = Crack Growth Rate, inches/cycle

ΔK = Stress Intensity Factor Range, $\text{Ksi}\sqrt{\text{in}}$

3. Assume there is a thin Surface defect oriented normal to the Maximum Surface Stress At the inside Surface of the thread root radius where the Maximum stress occurs.

4. Assume that the Stress Range is Equal to the Maximum Surface Stress.

Procedure given in Reference (2) will be followed:

1. The Fracture Toughness, K_{IC}, is:

$$K_{IC} = 100 \text{ ksi}\sqrt{in}$$

2. From Reference (1), the crack Growth Rate, da/dN, is:

$$\frac{da}{dN} = C_o \Delta K^n$$

$$\frac{da}{dN} = 0.66 \times 10^{-8} (\Delta K)^{2.25}$$

[For 4340 Mat'L from Ref. (1)]

Where: $\frac{da}{dN}$ = Crack Growth Rate, inches/cycle

C_o = Empirical intercept Constant

ΔK = Stress Intensity Factor Range, ksi$\sqrt{in}$

n = Slope of da/dN Versus Log ΔK Curve

BY DBP DATE 12/19/77 SUBJECT MACH 10 Heater Vessel SHEET NO 4 OF 5
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Procedure (continued)

The Crack Growth Rate Equation From Reference (1) is shown in the Curve below. Note that the Equation is an Upper bound of the plotted data.

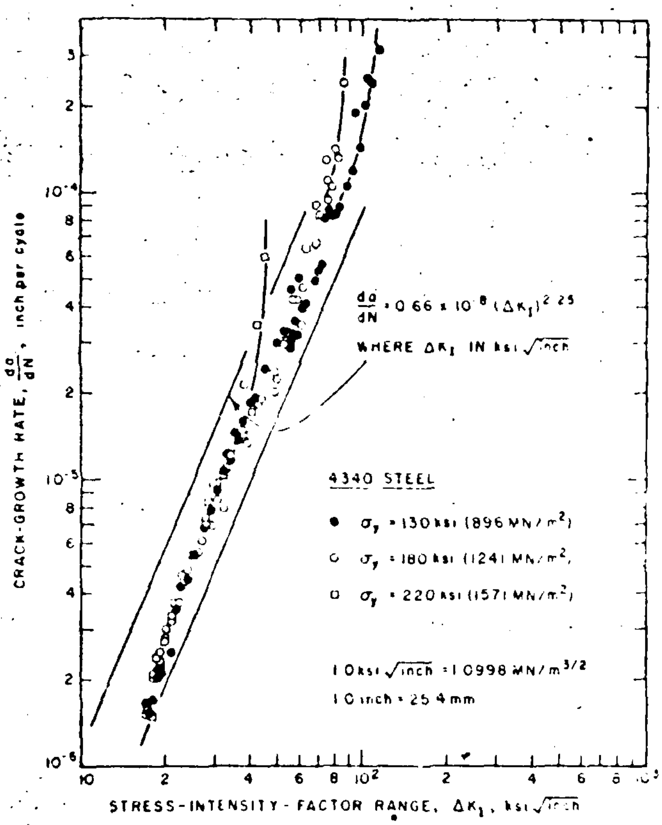

FIG. 9 - *Fatigue-crack growth in 4340 steel of various yield strengths*

Procedure (continued)

3. For a thick-walled Pressure Vessel containing a thin ($a/\ell \approx 0$) surface defect oriented normal to the Maximum surface stress, the critical crack depth, a_{cr}, is:

$$a_{cr} \cong \frac{K_c^2}{1.25\,\pi\,\sigma^2} \quad \{\text{Minimum } a_{cr}\}$$

where: a_{cr} = Critical Crack Depth, inches

K_c = Fracture Toughness, Ksi$\sqrt{in}$

σ = Maximum Surface Stress, Ksi

4. The Number of Cycles to grow to critical flaw size (failure), N, is:

$$N = \frac{2}{(n-2)\,C_o\,M^{n/2}\,\Delta\sigma^n}\left(\frac{1}{a_i^{(n-2)/2}} - \frac{1}{a_{cr}^{(n-2)/2}}\right)$$

Where: N = Number of Cycles to Failure

a_i = initial Crack Depth, inches

n = Slope of da/dN versus Log ΔK Curve

a_{cr} = Critical Crack Depth, inches

C_o = Empirical intercept constant for ΔK in psi$\sqrt{in}$

$\Delta\sigma$ = Applied cyclic stress Range, psi

$M = 1.25\,\pi$

Threads on Right End Closure

If $\sigma = \Delta\sigma = 229{,}708$ psi and $K_{IC} = 100$ Ksi$\sqrt{in}$

1. $K_{IC} = 100$ Ksi$\sqrt{in}$

2. Critical Crack Depth

$$a_{cr} = \frac{1}{1.25\pi}\left(\frac{100{,}000}{229{,}708}\right)^2 = 0.0482601''$$

3. Cycles to Failure

$$C_o = 1.173664411 \times 10^{-15} \text{ for } \Delta K \text{ in psi}\sqrt{in}$$

$$(n-2) = 0.25 \qquad M^{n/2} = (1.25\pi)^{1.125} = 4.659264564$$

$$\Delta\sigma^n = (229{,}708)^{2.25} = 1.155170983 \times 10^{12}$$

$$\frac{1}{a_{cr}^{(n-2)/2}} = \frac{1}{(0.0482601)^{0.125}} = 1.460667858$$

$$N = 1{,}266.433554\left[\frac{1}{a_i^{0.125}} - 1.460667858\right]$$

$$a_i = \left(\frac{1{,}266.433554}{N + 1{,}849.838787}\right)^8$$

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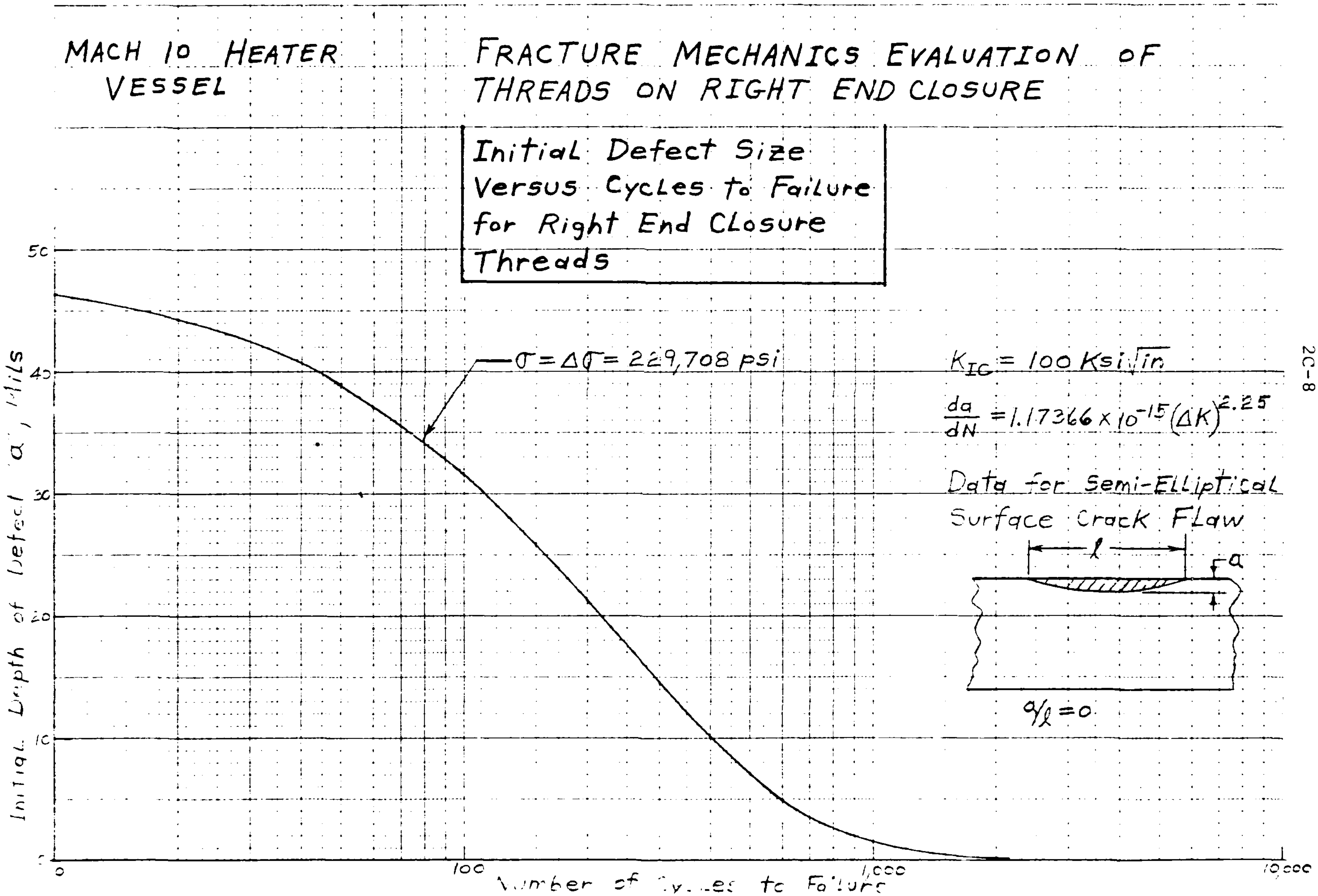
CHKD. BY DATE PROJ. NO JP1210

a_i Versus N for Threads on Right End Closure, $\sigma = \Delta\sigma = 229{,}708$ psi
$K_{IC} = 100\ Ksi\sqrt{in}$ –
ASTM A-723, CL. 2 Material

a_i inches	N Cycles
0.046223	10
0.044282	20
0.038987	50
0.031672	100
0.025864	150
0.021228	200
0.014501	300
0.010079	400
0.007118	500
0.001521	1,000
0.0001371	2,000
0.000001365	5,000

$$a_i = \left(\frac{1{,}266.433554}{N + 1{,}849.838787}\right)^8$$



MACH 10 HEATER VESSEL
FRACTURE MECHANICS EVALUATION OF THREADS ON RIGHT END CLOSURE
Initial Defect Size Versus Cycles to Failure for Right End Closure Threads
σ = Δσ = 229,708 psi
K_IC = 100 Ksi√in
da/dN = 1.17366 x 10^-15 (ΔK)^2.25
Data for Semi-Elliptical Surface Crack Flaw
ℓ
a
a/ℓ = 0
50
40
30
20
10
0
100
1,000
10,000
Initial Depth of Defect 'a', Mils

APPENDIX 3A

FATIGUE EVALUATION OF THREADS ON DOWNSTREAM END OF MACH 10 HEATER VESSEL

STRUCTURAL EVALUATION OF MACH 10 HEATER VESSEL/NOZZLE AREA

The downstream end of the M10 Heater Vessel and Nozzle area is shown on Drawing 77-F-1131. The design pressure for this area is 15,000 psi.

EVALUATION OF THREADED CLOSURES

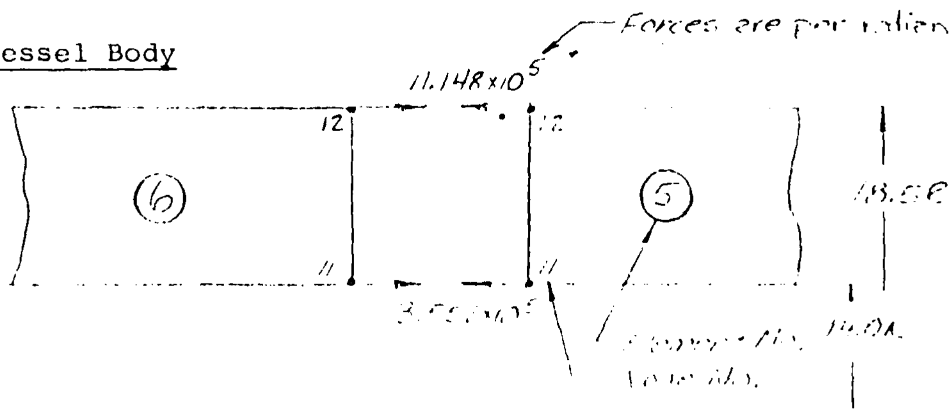
There are three major threaded closures in the M10 Heater/Nozzle assembly. These are: (1) Heater Body/Left End Main Nut, (2) Heater Body/Outer Housing, and (3) Nozzle Block/Piston Block. The external loading consists of 15,000 psi internal pressure up to the Left End Main Nut plus 4,000 psi preload pressure exerted at the piston block.

The first task was to determine the load paths in the assembly. This was accomplished by use of a coarse model of the entire assembly. The boundary conditions imposed were those before rupture of the diaphragms. The total axial pressure load exerted on the assembly is given by:

$$F_{TOT} = \pi(14)^2(15{,}000) = 9.236 \times 10^6 \text{ lbs}$$

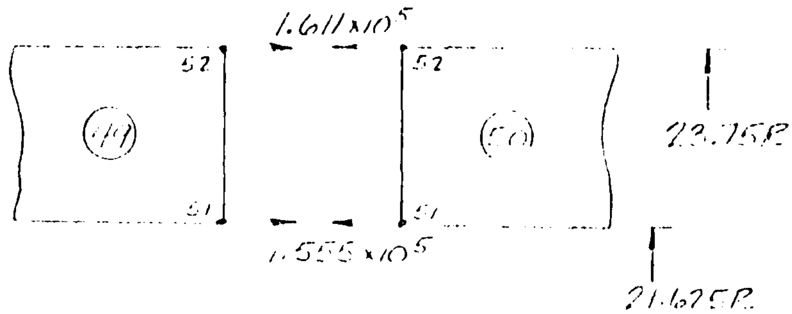
By taking various cuts through the model, the load being transmitted through the components can be determined. From Run ODAND4V (1/12/78):

Heater Vessel Body



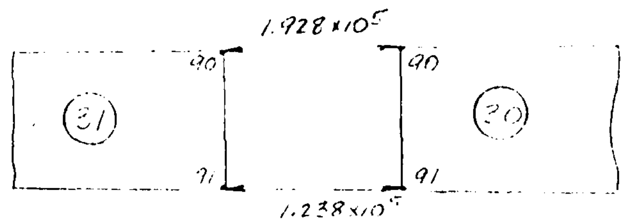
$$F = 2\pi(11.148 + 3.552) \times 10^5 = 9.236 \times 10^6 \text{ lbs (Tension)}$$

<u>Outer Housing</u>



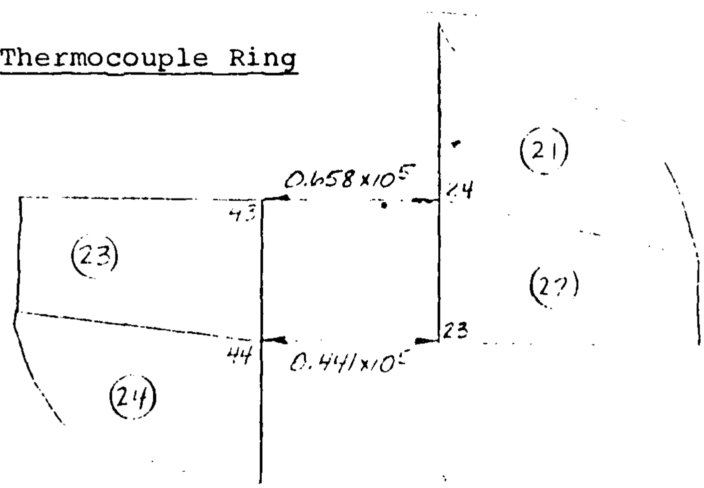
$$F = 2\pi(1.611 + 1.555) \times 10^5 = 1.989 \times 10^6 \text{ lbs (Tension)}$$

<u>Housing, Particle Separator</u>



$$F = 2\pi(1.928 + 1.238) \times 10^5 = 1.989 \times 10^6 \text{ lbs (Compression)}$$

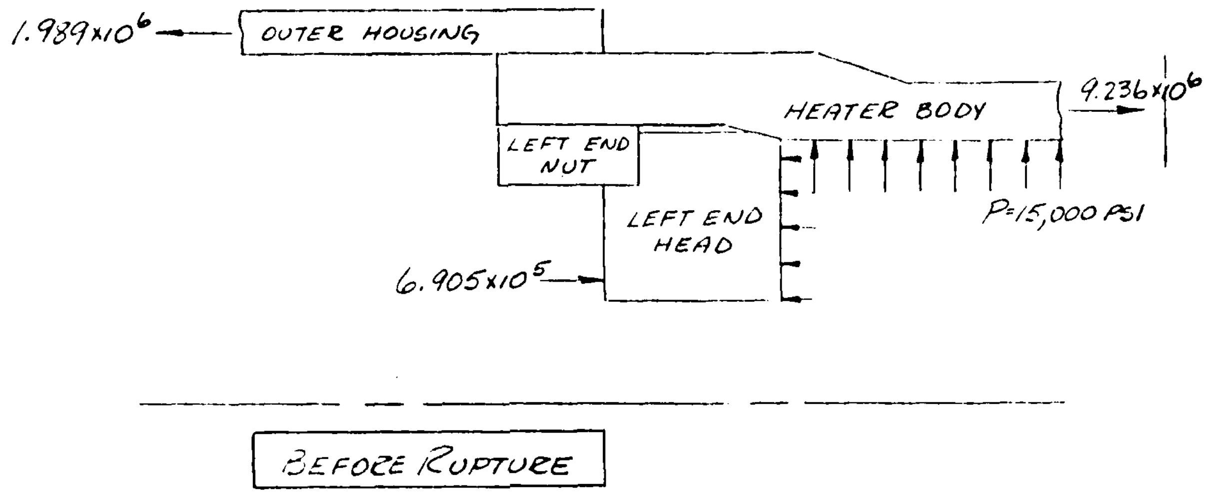
<u>Left End Head/Thermocouple Ring</u>



$$F = 2\pi(.441 + .658) \times 10^5 = 6.905 \times 10^5 \text{ lbs (Compression)}$$

THREADED CLOSURE - DOWNSTREAM END OF M10 HEATER

The configuration of the downstream end of the M10 Heater is shown below, along with the imposed loading obtained from the overall model.



The ANSYS finite element model for this area consists of 1985 Isoparametric (STIF42) elements. The threaded connections between the Heater Body and Outer Housing and between the Heater Body and Left End Nut are modeled by 27 element teeth. The nodes common between mating threads were coupled together if they were found to be in compression and let free if they were in tension. Only the normal direction was coupled, and the nodes were free to slide tangentially. No friction was assumed between threads.

The resulting isostress plots of the various components are shown in Figures 3A-1 through 3A-5. The maximum stresses occurring in each component (exclusive of threads) are listed below.

Ref. Run ODANDTX (2/3/78)

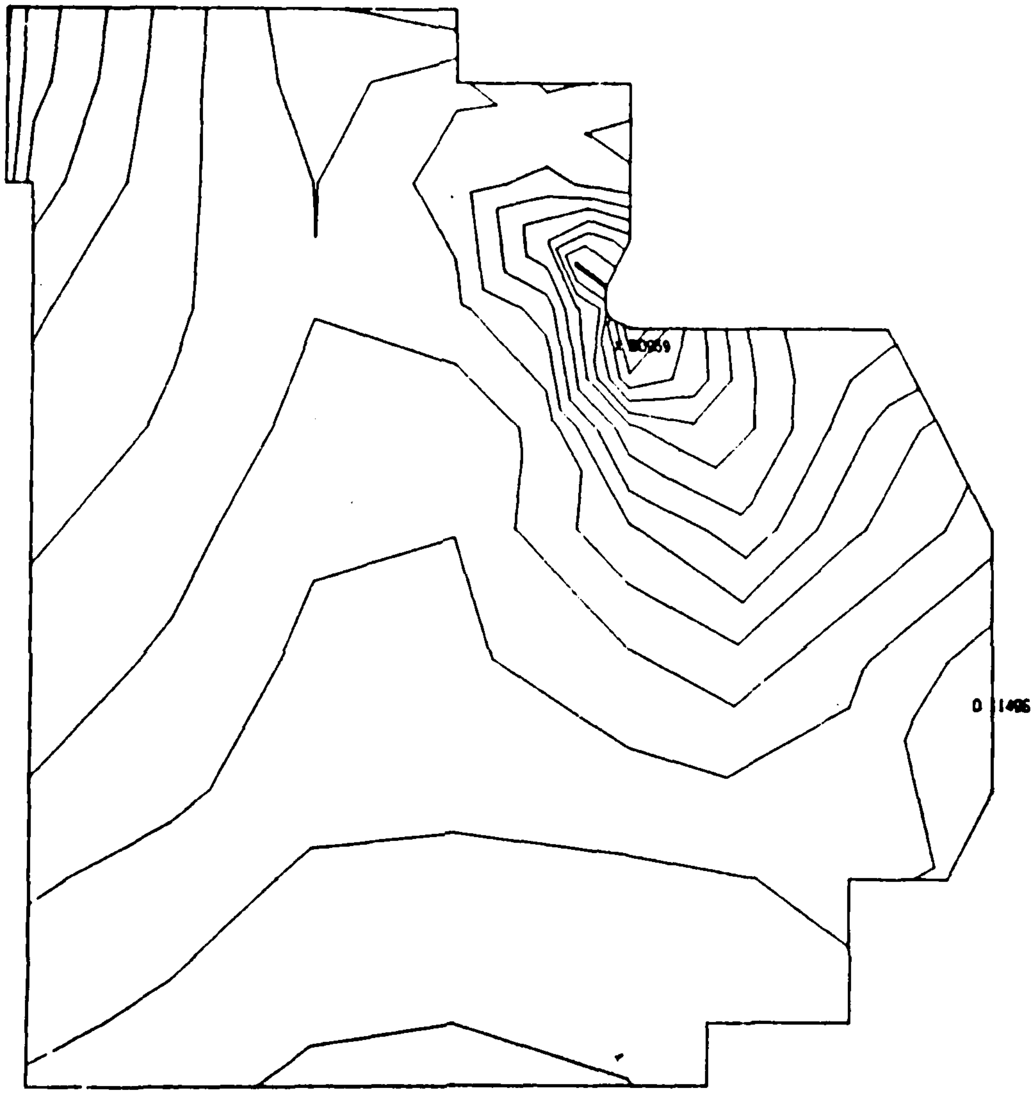
Component	Maximum Stress Intensity (psi)	Element
Heater Body	67,200	1021
Outer Housing	28,900	1716
Nut	78,000	251
Head	51,000	82

The distribution of forces along the thread interfaces is plotted in Figure 3A-6. The overall finite element model of the downstream end of the heater vessel does not have sufficient detail in the thread areas to adequately analyze a single tooth. This was accomplished by imposing the loading conditions (interface forces and boundary displacements) from the overall model onto a detailed finite element model of a single tooth. The most severely loaded tooth in each interface was analyzed.

The total interface force was converted to an equivalent pressure applied to the area of contact between the two teeth. The corresponding boundary displacements were linearly interpolated when necessary to obtain nodal displacements for the detailed model.

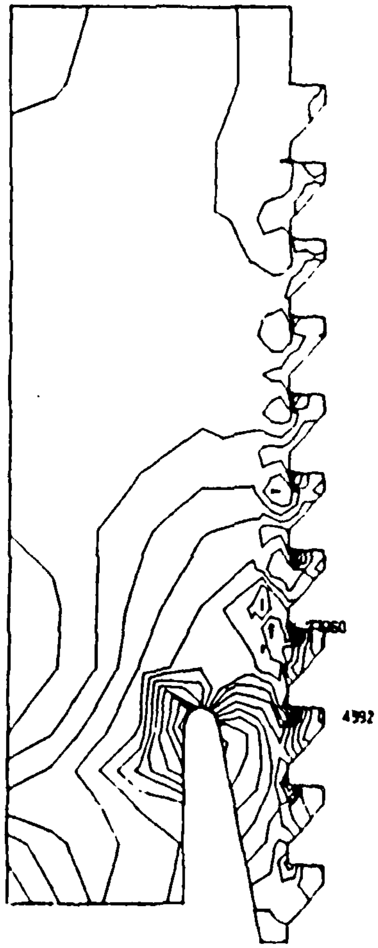
Figure 3A-7 shows the stress intensity isoclines for a typical tooth analyzed. The peak stress intensity in each tooth is listed below.

Component	σ_I (max) psi	Location	Ref. Run No.
Body/Nut Tooth No. 5	133,800	Surface of Element 103	ODANDGD
Body/Housing Tooth No. 6	49,400	Element 289	ODAND2E



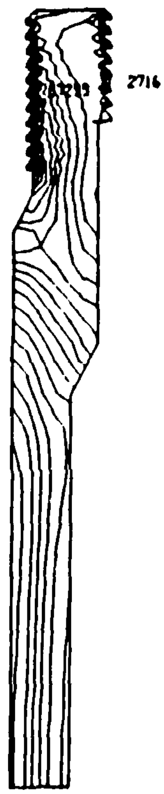
Left End Head

FIGURE 3A-1 - DOWNSTREAM END OF HEATER VESSEL
ISOSTRESS PLOT OF STRESS INTENSITY



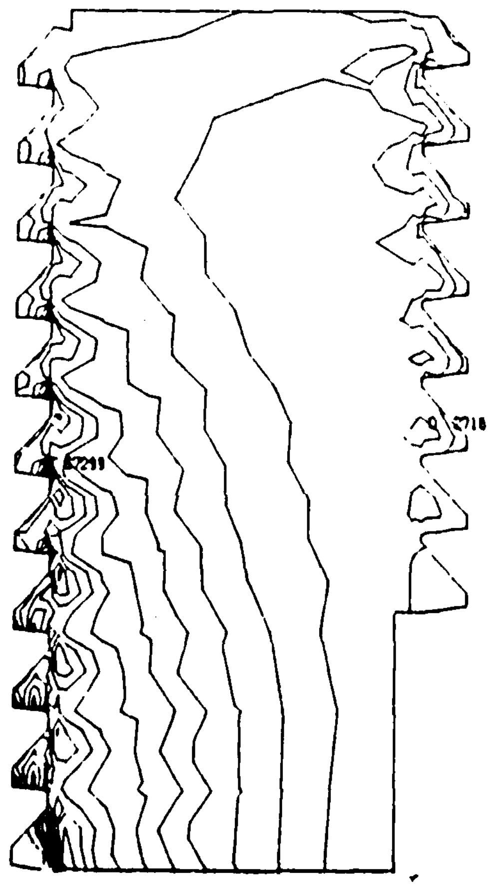
Left End Main Nut

FIGURE 3A-2 - DOWNSTREAM END OF HEATER VESSEL
ISOSTRESS PLOT OF STRESS INTENSITY



Heater Vessel Body

FIGURE 3A-3 - DOWNSTREAM END OF HEATER VESSEL
ISOSTRESS PLOT OF STRESS INTENSITY



Top of Heater Vessel Body

FIGURE 3A-4 - DOWNSTREAM END OF HEATER VESSEL
ISOSTRESS PLOT OF STRESS INTENSITY

Outer Housing

FIGURE 3A-5 - DOWNSTREAM END OF HEATER VESSEL
ISOSTRESS PLOT OF STRESS INTENSITY

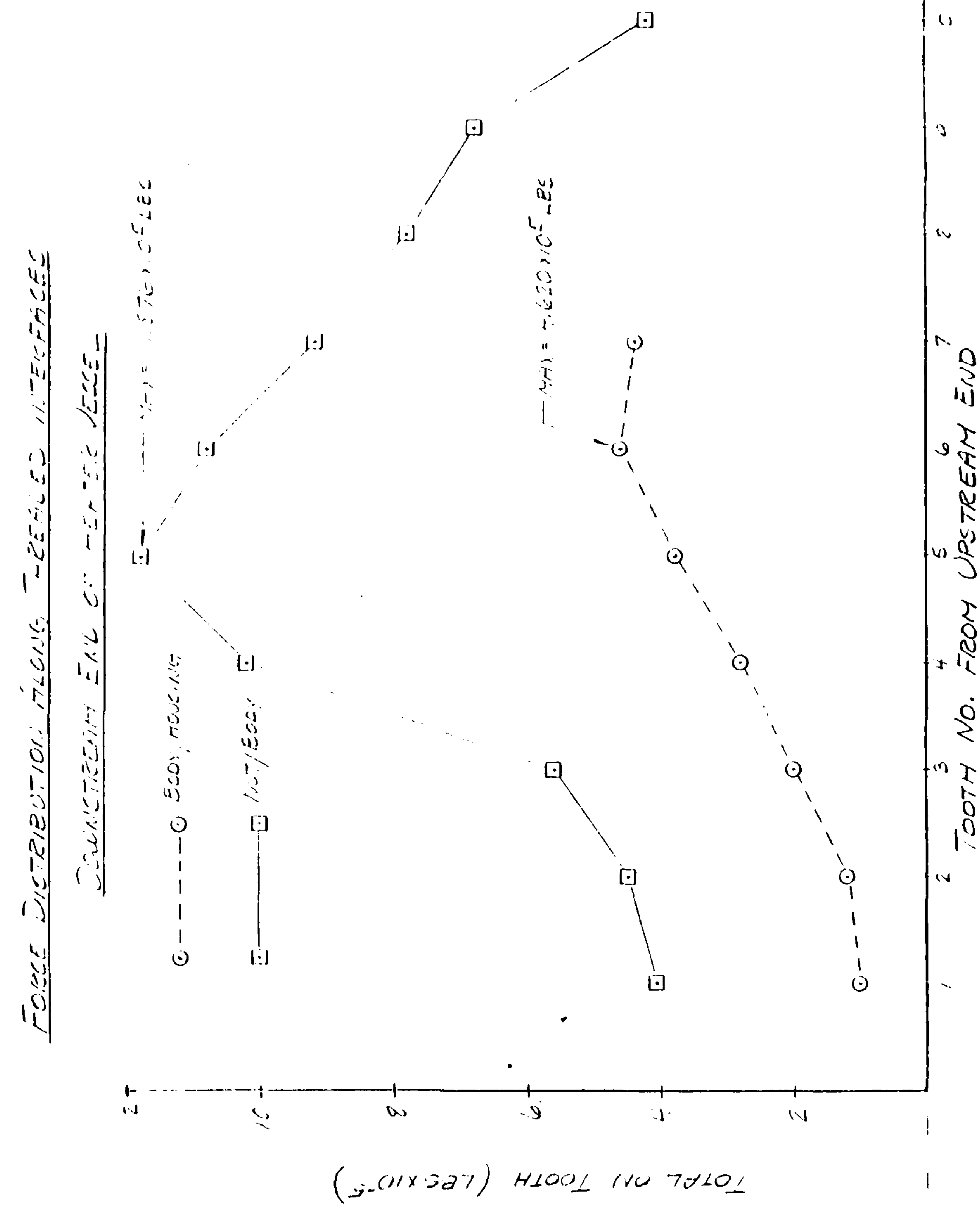


FIGURE 3A-6 - FORCE DISTRIBUTION ALONG THREADED INTERFACES
DOWNSTREAM END OF HEATER VESSEL

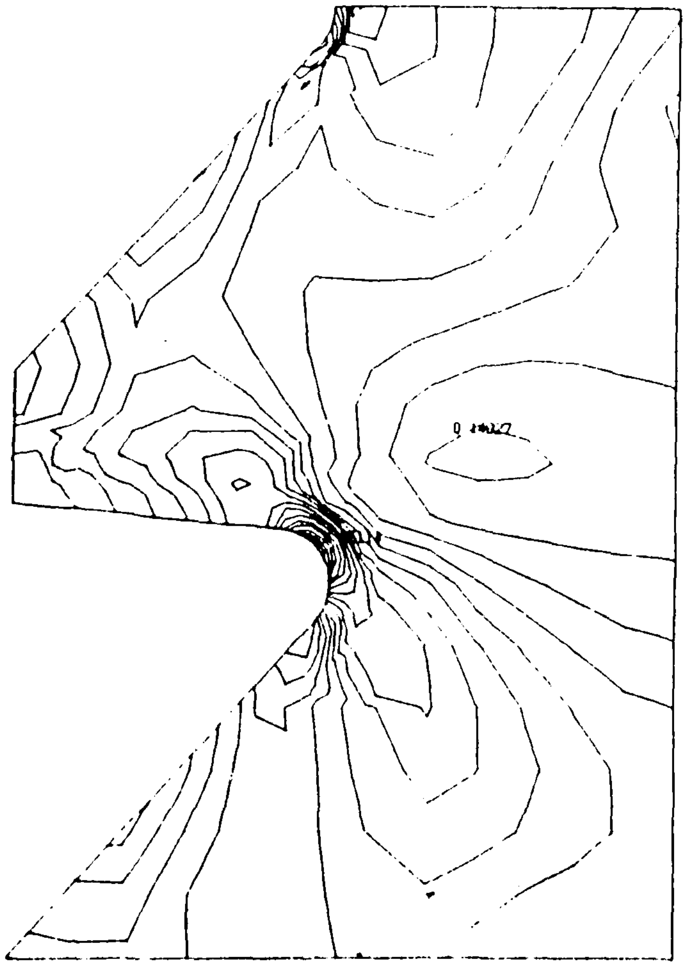


FIGURE 3A-7 - DOWNSTREAM END OF HEATER VESSEL
ISOSTRESS PLOT OF STRESS INTENSITY

FATIGUE ANALYSIS OF BUTTRESS TOOTH

The maximum stress intensity occurs in the 5th tooth of the M10 body/outer housing thread interface and is 133,800 psi at the surface of element 103 (root radius area).

The stress distribution across a section containing a cir-

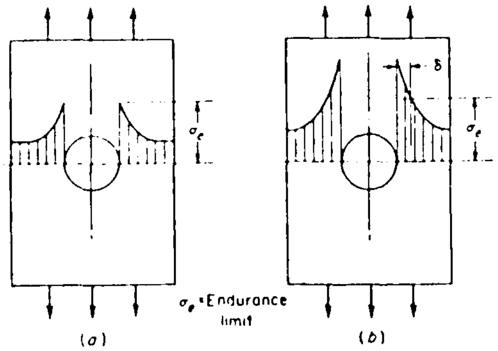


Fig. 6.29. Stress Distribution about a Circular Hole in a Bar

cular hole, Fig. 6.29a, has a high stress gradient at the edge of the hole. If the load is just sufficient to bring the peak stress up to the endurance limit, a fatigue failure would hardly be expected since the volume of material at this stress is zero. A finite volume of material must be at the endurance limit before a crack will form, and to obtain this volume of material the endurance limit stress must exist at some finite depth, δ, below the surface; therefore, the steeper the stress gradient, the higher the load required to produce fatigue failure, Fig. 6.29b.

The dimension, δ, is a property of the material; and, in general, hard, fine-grained materials have small values of δ, whereas soft, coarse-grained materials have larger values. The relationship between δ and steel tensile strengths, based on correlating fatigue data and the shear theory of failure is shown in Fig. 6.30.

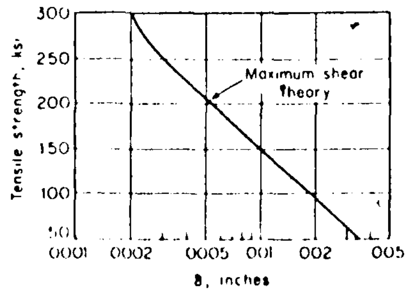


Fig. 6.30. Material Constant δ vs. Tensile Strength for Steel

For the M10 Heater Body, the tensile strength is 135,000 psi, which gives a "δ" of 0.00125 in.

We must, therefore, compute the stress intensity in the area of the root radius at a depth of 1.25×10^{-3} in. It will be assumed that the stress distribution in the vicinity of the root radius is the same as that around a small hole in the middle of a flat plate subjected to uniform tension. From "Theory of Elasticity," Timoshenko and Goodier, 2nd Edition, page 78, the stress distribution around the hole is given by:

$$\sigma_r = \frac{S}{2}\left[1 - \left(\frac{a}{r}\right)^2\right] + \frac{S}{2}\left[1 + 3\left(\frac{a}{r}\right)^4 - 4\left(\frac{a}{r}\right)^2\right]\cos 2\theta$$

$$\sigma_\theta = \frac{S}{2}\left[1 + \left(\frac{a}{r}\right)^2\right] - \frac{S}{2}\left[1 + 3\left(\frac{a}{r}\right)^4\right]\cos 2\theta$$

$$\tau_{r\theta} = -\frac{S}{2}\left[1 - 3\left(\frac{a}{r}\right)^4 + 2\left(\frac{a}{r}\right)^2\right]\sin 2\theta$$

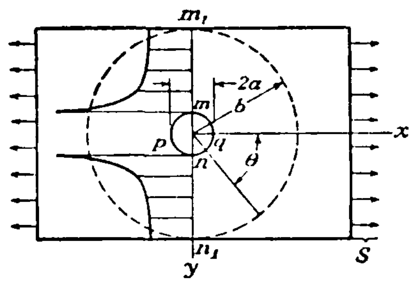


FIG. 48.

When $\theta = 0°$, $\tau_{r\theta} = 0$ and the principle stresses are:

$$\sigma_r = \frac{S}{2}\left[2 + 3\left(\frac{a}{r}\right)^4 - 5\left(\frac{a}{r}\right)^2\right]$$

$$\sigma_\theta = \frac{S}{2}\left[-3\left(\frac{a}{r}\right)^4 + \left(\frac{a}{r}\right)^2\right]$$

And the stress intensity is:

$$\sigma_I = S\left[1 + 3\left(\frac{a}{r}\right)^4 - 3\left(\frac{a}{r}\right)^2\right]$$

Therefore, the assumed distribution in the vicinity of the thread root radius is:

$$\sigma_I = S\left[1 + A\left(\frac{a}{r}\right)^4 - B\left(\frac{a}{r}\right)^2\right]$$

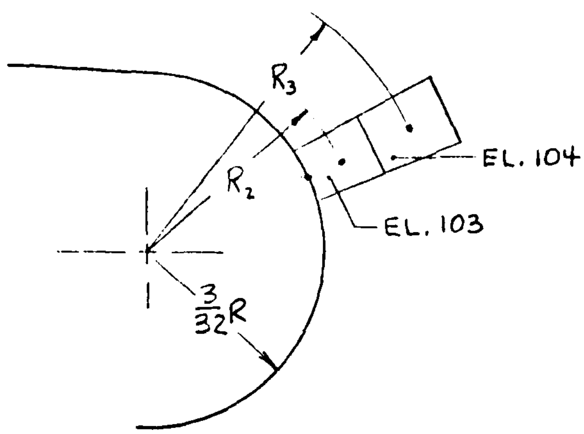
where: a = thread root radius = 0.09375 in.

$r = a + \delta$

δ = distance from surface, in.

S,A,B = constants to be determined

If the stress intensity at three points in the area of interest are known, S, A and B can be determined. From Run ODANDTX:

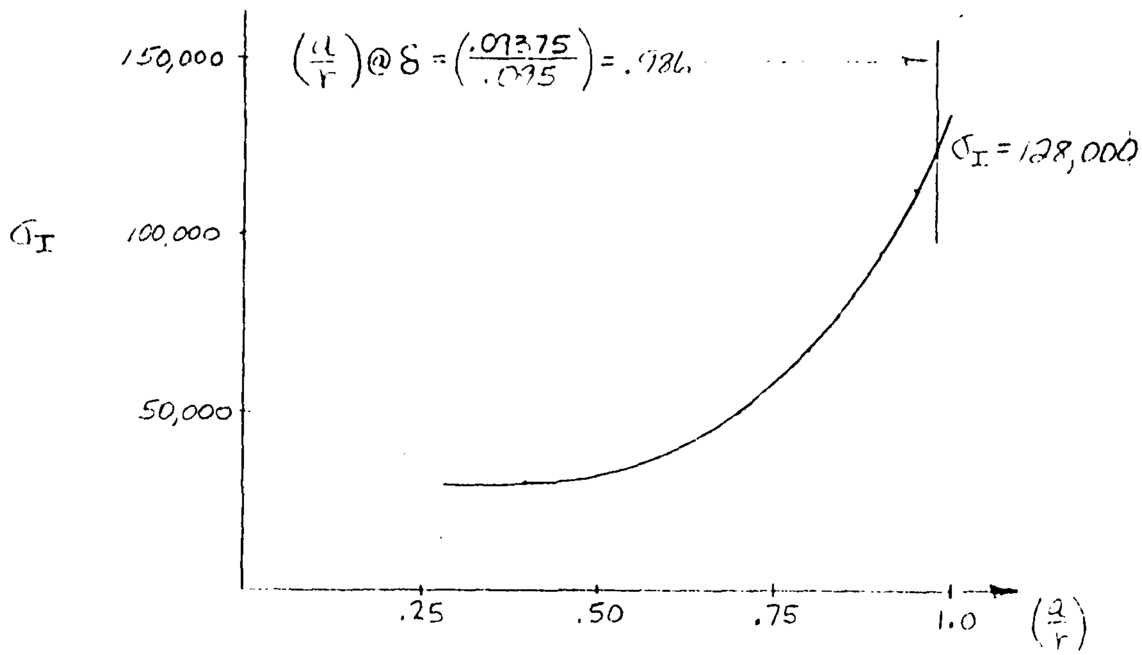


r (IN)	σ_I (PSI)
.09375	133,800
.10809	85,100
.15026	41,100

Solving from S, A and B from the above yields:

S = 30,022; A = 4.1075; B = .6502

A plot of this equation is shown below:



Therefore, for the fatigue analysis, the maximum stress intensity is:

$$\sigma_I = 128,000 \text{ psi}$$

The stress intensity range for one pressure cycle is:

$$\sigma_{RANGE} = 128,000 \text{ psi}$$

$\sigma_{ALT} = 64,000$ psi $\qquad \sigma_y = 120,000$ psi

$\sigma_{MEAN} = 64,000$ psi $\qquad \sigma_u = 135,000$ psi

The following procedure for accounting for the effects of mean stress is from:

Snow, A. L. and Langer, B. F., "Low Cycle Fatigue of Large Diameter Bolts," ASME J. of Engrg. for Industry, Feb. 1967.

$$\sigma_{ALT} + \sigma_{MEAN} = 128{,}000$$

Since $\sigma_{ALT} < \sigma_y$ and $\sigma_{ALT} + \sigma_{MEAN} > \sigma_y$,

$$\sigma_{MEAN} = \sigma_y - \sigma_{ALT} = 120{,}000 - 64{,}000 = 56{,}000 \text{ psi}$$

$$\sigma_{eq} = \frac{7\sigma_{ALT}}{8 - \left[1 + \frac{\sigma_{MEAN}}{\sigma_u}\right]^3} = \frac{(7)(64{,}000)}{8 - \left[1 + \left(\frac{56{,}000}{135{,}000}\right)\right]^3}$$

$$\sigma_{eq} = 86{,}700 \text{ psi}$$

This equivalent stress will be used to enter the fatigue curve, Figure 3A-7A. This curve is from ASME Paper No. 76-PVP-62. Since the theoretical fatigue curves from this paper were obtained on small polished specimens tested in air, factors must be applied to account for size effects and scatter in the date. Therefore, a factor of either 2 on stress or 20 on cycles, whichever is more conservative at each point, was applied to the mean failure curve to obtain a design curve which accounts for these effects. The Design Life for a σ_{eq} of 86,700 psi is:

<u>N = 1,900 cycles</u>

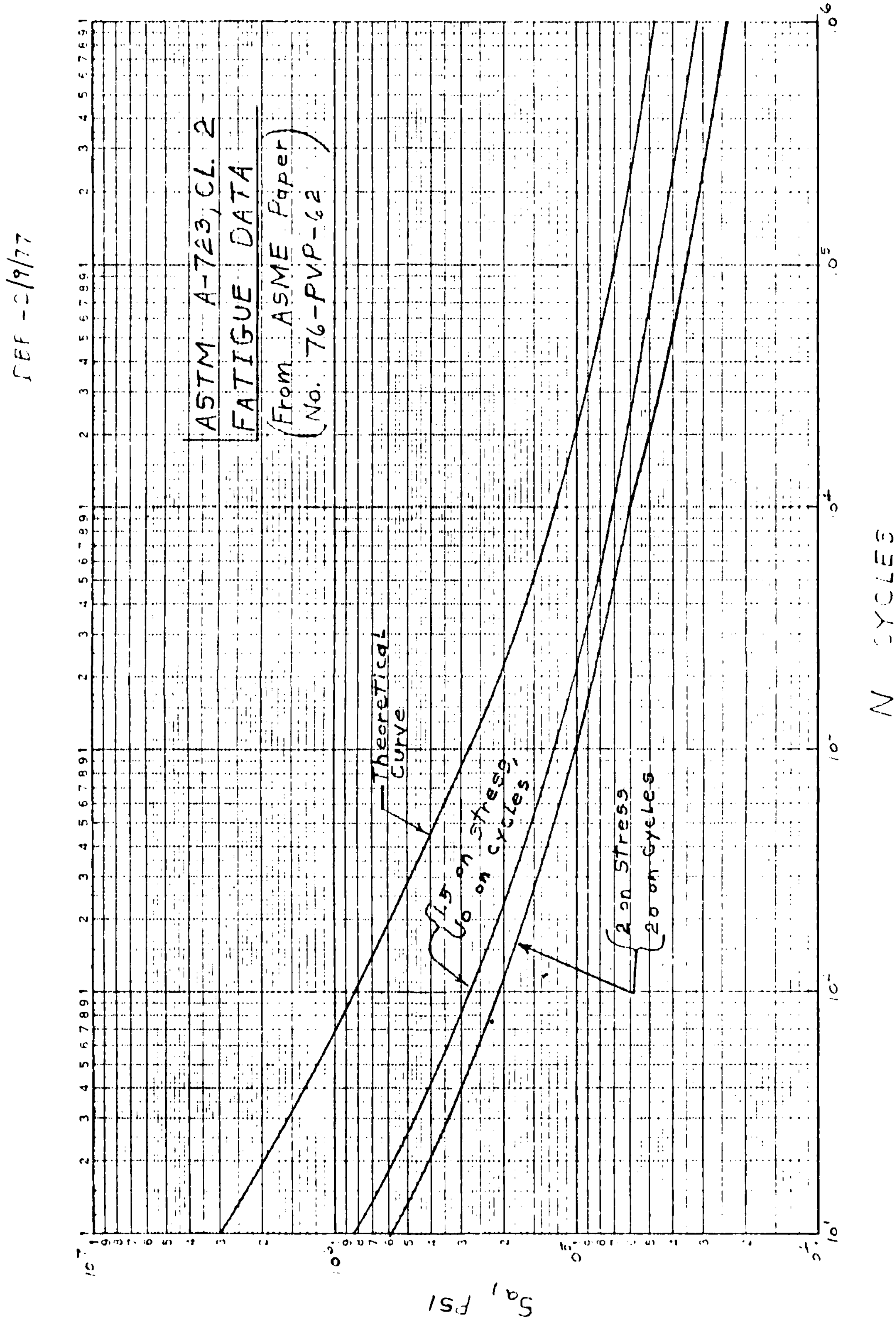
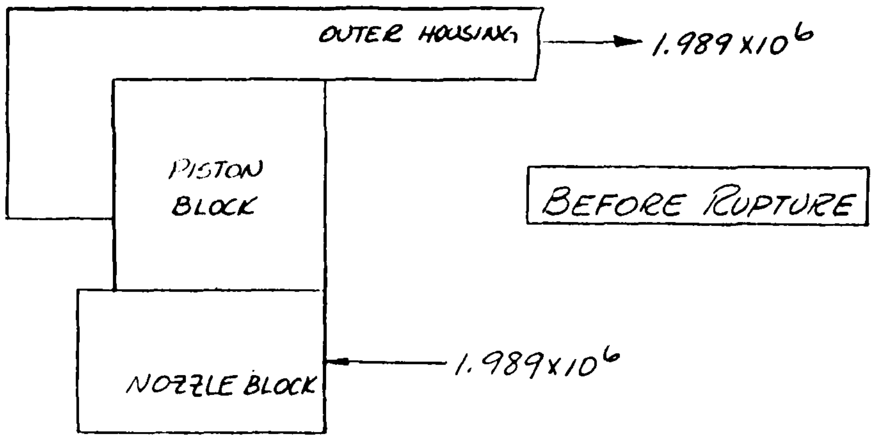

ASTM A-723, CL. 2
FATIGUE DATA
(From ASME Paper No. 76-PVP-62)
Theoretical Curve
1.5 on Stress, 10 on Cycles
2.0 Stress, 20 on Cycles
N CYCLES
Sa, PSI

FIGURE 3A-7A

THREADED CLOSURE - M10 PISTON BLOCK/NOZZLE BLOCK

The configuration of the M10 Piston Block/Nozzle Block threaded closure is shown below, along with the imposed loading obtained from the overall model.



The ANSYS finite element model of this area consists of 1027 Isoparametric (STIF42) elements. The method of handling the threaded closure is the same as for the downstream M10 Heater model. The Nozzle block contains 8 - 2" diameter holes on a 17-1/2" diameter bolt circle. To account for the increased flexibility of nozzle block due to these holes, the modulus of elasticity, E, was adjusted as follows:

$$E_{MOD} = \frac{\text{Area excluding holes}}{\text{Area including holes}} \times 30 \times 10^6 \text{ psi}$$

$$E_{MOD} = (0.7715)(30 \times 10^6) = 23.14 \times 10^6 \text{ psi}$$

This modified E was used for those elements of the nozzle block which are within the annulus formed by the holes. To account for the resistance to rotation imposed upon the nozzle block by

the nozzle throat insert carrier, all nodes along the inboard surface of the nozzle block were required to have the same radial displacement.

The resulting isostress plots of the various components are shown in Figures 3A-8 through 3A-10. The maximum stresses occurring in each component (exclusive of threads) are listed below.

Ref. ODANDM1 (1/23/78)

Component	Maximum Stress Intensity (psi)	Element
Nozzle Block	21,700	181
Piston Block	25,100	834
Outer Housing	15,300	930

The distribution of forces along the thread interface is shown in Figure 3A-11. Again, a detailed model of the Piston Block tooth #4 was used to determine the stress state in the tooth. The method followed was identical to that used in the previous section. The maximum stress intensity occurs at the surface of element 135 and is 23,600 psi (Ref. ODANDXA - 2/27/78).

Even though tooth #4 is the most highly loaded, Figure 3A-9 shows that the maximum stress intensity occurs at the last tooth (#9) and is greater (25,100 psi vs. 23,600) than that obtained from the detail tooth model. Figure 3A-12 shows that the piston block and outer housing are undergoing rotations which will induce large hoop forces at the upper end of these components. This increase in hoop loading is the major factor contributing to the larger stress intensity in the last tooth. As a result, the interfacial loadings and boundary displacements from the overall model for the last tooth were imposed upon the detail tooth model. The maximum stress intensity from this case (Ref. ODANDKJ - 2/28/78) is 24,200 psi at the surface of

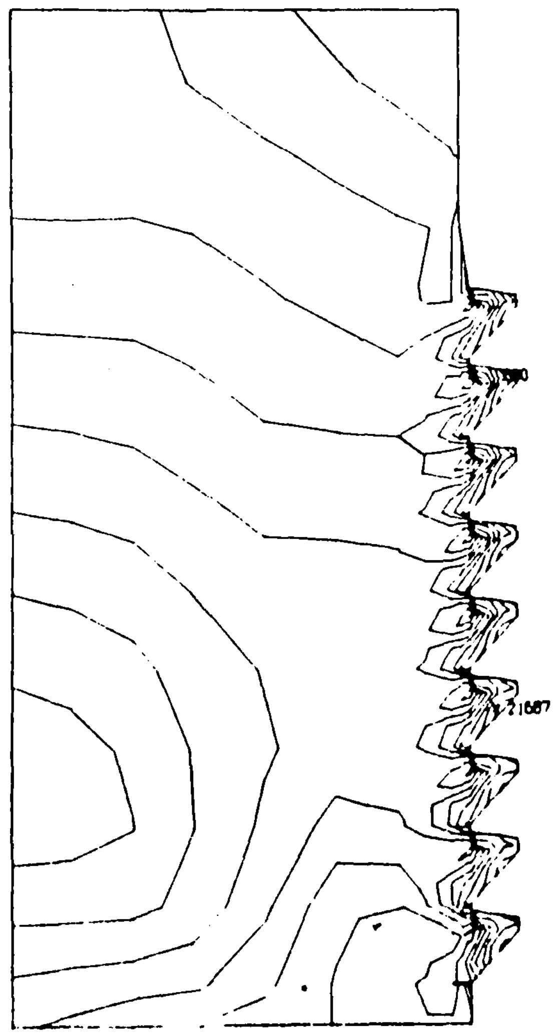


FIGURE 3A-8 - M10 NOZZLE BLOCK/PISTON BLOCK CLOSURE
ISOSTRESS PLOT OF STRESS INTENSITY

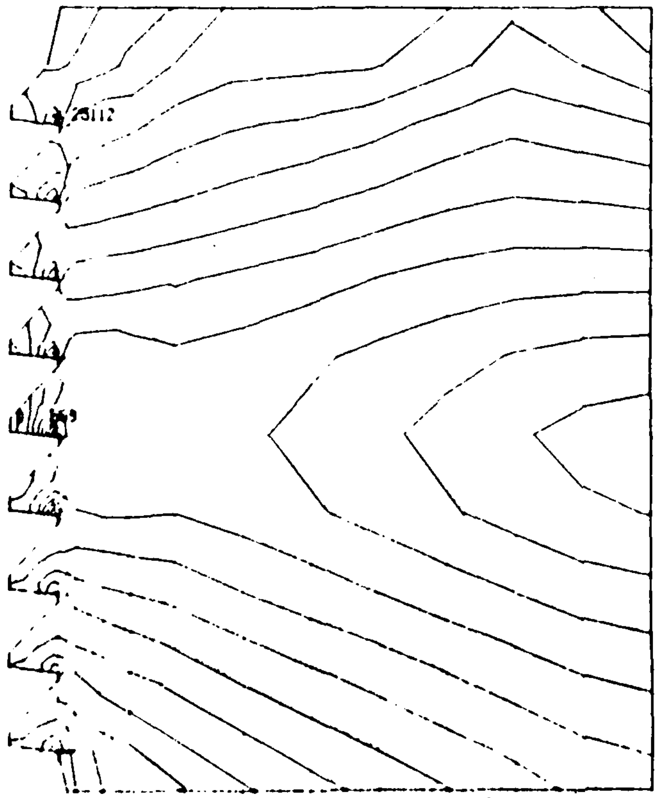


FIGURE 3A-9 - M10 NOZZLE BLOCK/PISTON BLOCK CLOSURE
ISOSTRESS PLOT OF STRESS INTENSITY

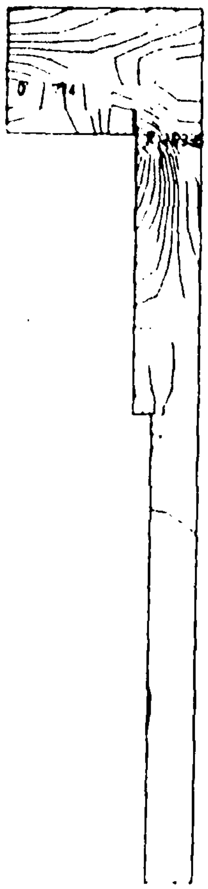


FIGURE 3A-10 - M10 NOZZLE BLOCK/PISTON BLOCK CLOSURE
ISOSTRESS PLOT OF STRESS INTENSITY

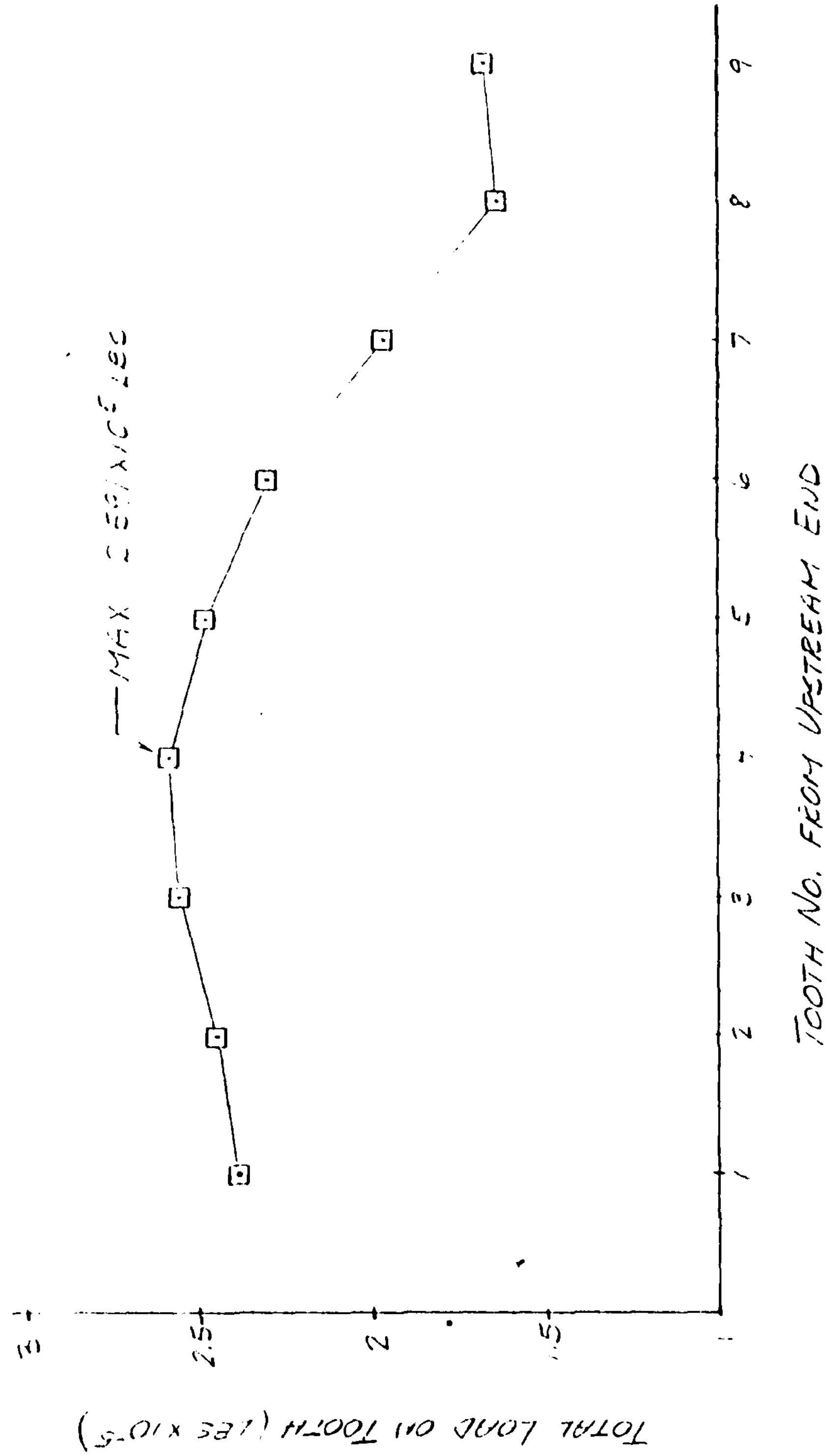


FIGURE 3A-11 - FORCE DISTRIBUTION ALONG THREADED
INTERFACE PISTON BLOCK/NOZZLE BLOCK

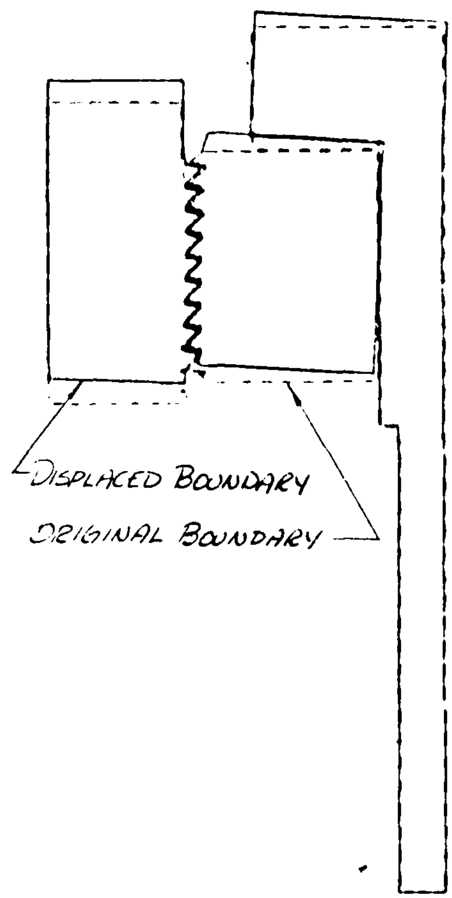
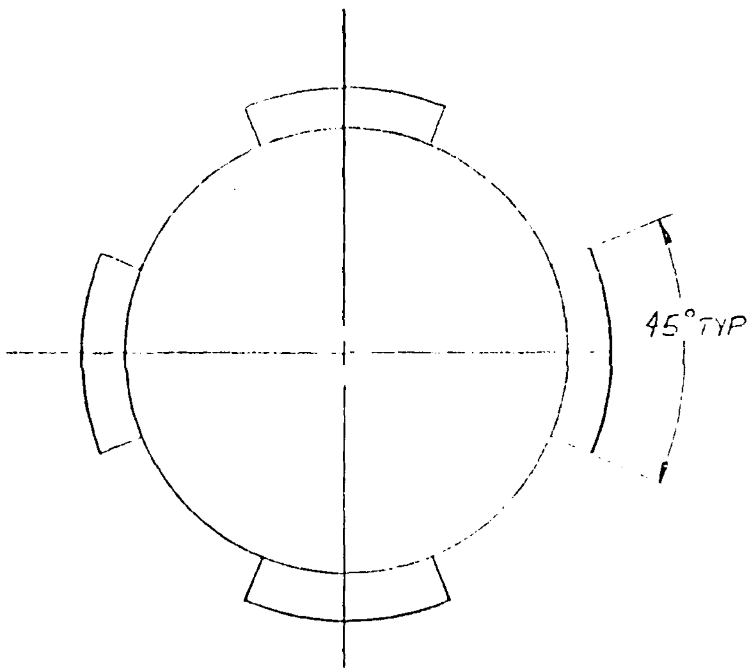


FIGURE 3A-12 - BOUNDARY DISPLACEMENTS -
M10 NOZZLE BLOCK/PISTON BLOCK CLOSURE
(Ref. ODANDM1)

element 135. This value itself is less than the 25,100 psi value obtained from the overall model. The purpose of the more detailed thread model was to account for any localized stress concentration factors in the tooth profile. However, the primary loading on this thread is hoop stress which will not localize itself. For purposes of the fatigue analysis, the value of 25,100 psi will be used.

The threads on the M10 Piston Block are interrupted as shown below:



Percentage which is thread -- $\frac{(4)(45)}{360} = \approx 50\%$

Therefore, the stress intensity must be increased by a factor of 2, i.e.,

$$(\sigma_I)_{MAX} = 50,200 \text{ psi}$$

Following the procedure outlined in Section

$$\sigma_{ALT} = 25{,}100; \quad \sigma_{MEAN} = 25{,}100$$

$$\sigma_{eq} = \frac{(7)(25{,}100)}{8 - \left[1 + \frac{25{,}100}{135{,}000}\right]^3} = 27{,}700 \text{ psi}$$

which relates to a design life from Figure 3A-7A of

N = 400,000 cycles

APPENDIX 3B

FRACTURE MECHANICS EVALUATION OF THREADS ON DOWNSTREAM END OF MACH 10 HEATER VESSEL

The procedure followed herein is outlined in detail in Appendix C.

The thread material is modified AISI 4340, or "gun" steel (ASTM A-723, Class 2), with the following material properties:

$$\sigma_u = 135{,}000 \text{ psi}$$

$$\sigma_y = 120{,}000 \text{ psi}$$

$$K_{IC} = 100 \text{ KSI } \sqrt{\text{in.}}$$

From the stress analysis of the detailed tooth model,

$$\sigma_{MAX} = 133{,}800 \text{ psi}$$

The critical crack depth is:

$$a_{CR} = \frac{1}{1.25\pi}\left(\frac{100{,}000}{133{,}800}\right)^2 = 0.142 \text{ in.}$$

The cycles to failure is determined from:

$$C_o = 1.1737 \times 10^{-15} \text{ for } \Delta K \text{ in psi } \sqrt{\text{in.}}$$

$$(n - 2) = 0.25 \; M^{N/2} = (1.25\pi)^{1.125} = 4.6593$$

$$(\Delta\sigma)^n = (133{,}800)^{2.25} = 3.4239 \times 10^{11}$$

$$\frac{1}{a_{CR}^{(n-2)/2}} = \frac{1}{(.142)^{0.125}} = 1.2763$$

$$N = \text{cycles to failure} = \frac{2}{(n-2)C_o M^{N/2} \Delta\sigma^n} \left(\frac{1}{a_i^{(n-2)/2}} - \frac{1}{a_{CR}^{(n-2)/2}} \right)$$

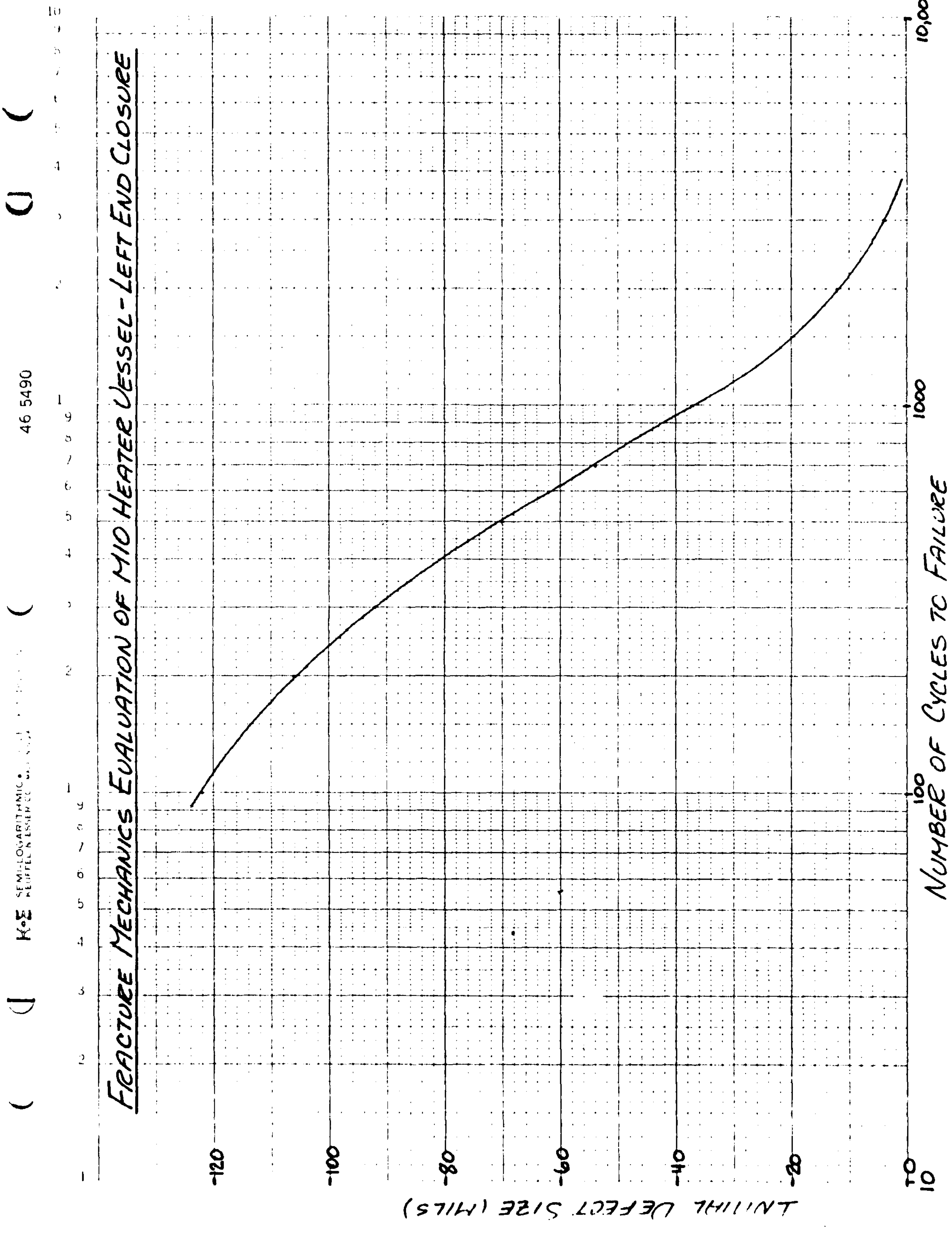
$$N = \frac{2}{(.25)(1.1737 \times 10^{-15})(4.6593)(3.4239 \times 10^{11})} \left[\frac{1}{a_i^{0.125}} - 1.2763 \right]$$

$$N = 4272.6 \left[\frac{1}{a_i^{0.125}} - 1.2763 \right]$$

From which:

$$a_i = \left[\frac{4272.6}{N + 5453.1} \right]^8$$

We now have a relationship for determining the cycles to failure for various defect sizes. This expression is shown in Figure 3B-1.


FRACTURE MECHANICS EVALUATION OF M10 HEATER VESSEL-LEFT END CLOSURE
NUMBER OF CYCLES TO FAILURE
INITIAL DEFECT SIZE (MILS)
10
100
1000
10,000
0
-20
-40
-60
-80
-100
-120
K+E SEMI-LOGARITHMIC
46 5490

APPENDIX 4A

PRIMARY STRESS EVALUATION
FOR
MACH 14/18 HEATER VESSEL

1. Primary Stresses in Cylinder and Liner

The primary stresses in the cylinder and liner section of the MACH 14/18 Heater Vessel due to an internal pressure of 46,000 psi and a shrink fit of 0.017" on the radius between the liner and the cylinder were calculated using a special-purpose computer program. The resulting stresses are listed and compared to the allowable stresses on page 4A- 2

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BY I.E.P. DATE 6/1/78 SUBJECT MACH 14/18 Heater Vessel SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO J7176

REF: ØDANDGD - 6/29/78

Liner Stresses Compared to the Allowable Stresses

Stress Category	Calculated Stress (psi)	Allowable Stress (psi)
P_m	95,568	$S_m = 80{,}000$
$P_m + P_b$	121,560	$1.5S_m = 120{,}000$

Stresses in Liner Are Due to Internal Pressure of 46,000 psi and Shrink Fit of 0.017" on Radius

$S_U = 160{,}000$ psi (Assumed)

$S_m = \frac{S_U}{2} = 80{,}000$ psi

Cylinder Body Stresses Compared to the Allowable Stresses

Stress Category	Calculated Stress (psi)	Allowable Stress (psi)
P_m	82,886	$S_m = 72{,}500$
$P_m + P_b$	107,086	$1.5S_m = 108{,}750$

Stresses in Cylinder Body Are Due to Internal Pressure of 46,000 psi and Shrink Fit of 0.017" on Radius

$S_U = 145{,}000$ psi, $S_Y = 130{,}000$ psi for Cylinder Body

$S_m = \frac{S_U}{2} = 72{,}500$ psi

2. MAXIMUM STRESSES IN LINER AND CYLINDER BODY

The maximum stress intensities in the liner and cylinder body due to an internal pressure of 46,000 psi and a shrink fit of 0.017" on the radius between the liner and cylinder body were calculated by hand. These hand calculations are given on the following pages. The resulting stresses are summarized in two tables on page 4A-7.

Maximum Stresses in Liner And Cylinder Body

Reference: Strength of Materials, Part II, Timoshenko, pp. 211-214.

$$\sigma_t = \frac{a^2 p_i}{b^2 - a^2}\left(1 + \frac{b^2}{r^2}\right) \quad \left\{\text{Tangential or Hoop Stress}\right)$$

$$\sigma_r = \frac{a^2 p_i}{b^2 - a^2}\left(1 - \frac{b^2}{r^2}\right) \quad \left\{\text{Radial stress}\right)$$

1. Pressure Stresses

$a = 12''$ $b = 20''$ $p_i = 46,000$ psi

(a) At Inside Surface of Liner ($r = 12''$)

$$\sigma_t = \frac{(12)^2(46,000)}{(20)^2 - (12)^2}\left[1 + \left(\frac{20}{12}\right)^2\right]$$

$$\sigma_t = 25,875\left[1 + \left(\frac{20}{12}\right)^2\right] = 97,750 \text{ psi}$$

$$\sigma_r = -p_i = -46,000 \text{ psi}$$

$$S = \sigma_t - \sigma_r = 143,750 \text{ psi} \quad \left\{\text{Stress Intensity}\right)$$

(b) At Inside Surface of Cylinder ($r = 15.5''$)

$$\sigma_t = 25,875\left[1 + \left(\frac{20}{15.5}\right)^2\right] = 68,955 \text{ psi}$$

$$\sigma_r = 25,875\left[1 - \left(\frac{20}{15.5}\right)^2\right] = -17,205 \text{ psi}$$

$$S = \sigma_t - \sigma_r = 86,160 \text{ psi}$$

Maximum Stresses in Liner And Cylinder Body (continued)

2. Shrink Fit Stresses

$a = 12''$ $b = 15.5''$ $c = 20''$

$\delta = 0.017''$ $E = 30 \times 10^6$ psi

(a) Shrink Fit Pressure

$$p = \frac{E\delta(b^2-a^2)(c^2-b^2)}{2b^3(c^2-a^2)}$$

$$p = \frac{(30\times10^6)(0.017)\left[(15.5)^2-(12)^2\right]\left[(20)^2-(15.5)^2\right]}{2(15.5)^3\left[(20)^2-(12)^2\right]}$$

$p = 4,113$ psi

(b) At Inside Surface of Liner $(r = 12'')$

$$\sigma_t = -\frac{2pb^2}{b^2-a^2} = -\frac{2(4,113)(15.5)^2}{(15.5)^2-(12)^2} = -20,533 \text{ psi}$$

$\sigma_r = 0$

(c) At Inside Surface of Cylinder $(r = 15.5'')$

$$\sigma_t = \frac{p(b^2+c^2)}{c^2-b^2} = \frac{(4,113)\left[(15.5)^2+(20)^2\right]}{(20)^2-(15.5)^2}$$

$\sigma_t = 16,484$ psi

$\sigma_r = -p = -4,113$ psi

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BY DBP DATE 6/29/78 SUBJECT MACH 14/18 Heater Vessel SHEET NO 3 OF 4
CHKD BY DATE PROJ. NO JT1270

Maximum Stresses in Liner And Cylinder Body (continued)

3. Pressure Plus Shrink Fit Stresses

(a) At Inside Surface of Liner (r=12")

$\sigma_t = 97,750 - 20,533 = 77,217$ psi

$\sigma_r = -46,000 + 0 = -46,000$ psi

$S = \sigma_t - \sigma_r = 123,217$ psi

(b) At Inside Surface of Cylinder (r=15.5")

$\sigma_t = 68,955 + 16,484 = 85,439$ psi

$\sigma_r = -17,205 - 4,113 = -21,318$ psi

$S = \sigma_t - \sigma_r = 106,757$ psi

Stresses in Liner and Cylinder Body Due to 46,000 psi Internal Pressure Only

	At Inner Surface of Liner	At Inner Surface of Cylinder
σ_t Hoop Stress (psi)	97,750	68,955
σ_r Radial Stress (psi)	−46,000	−17,205
S Stress Intensity (psi)	143,750	86,160

Stresses in Liner and Cylinder Body Due to 46,000 psi Internal Pressure Plus 0.017" Shrink Fit

	At Inner Surface of Liner	At Inner Surface of Cylinder
σ_t Hoop Stress (psi)	77,217	85,439
σ_r Radial Stress (psi)	−46,000	−21,318
S Stress Intensity (psi)	123,217	106,757

APPENDIX 4B

FATIGUE EVALUATION OF THREADS
for
MACH 14/18 HEATER VESSEL
ORIGINAL DESIGN

FATIGUE EVALUATION OF THREADS

The fatigue analysis calculations used to calculate the fatigue design life of the threads are given on the following pages. The calculations are divided into the following parts:

(a) Summary of Loads on Main Cylinder Threads

(b) Equivalent Pressure Calculation for Maximum Thread Load on Bottom End

The first part of this appendix deals with the bottom end of the heater, while the second half concerns the outlet end.

AD-A079 317 O'DONNELL AND ASSOCIATES INC PITTSBURGH PA F/G 14/2
HYPERVELOCITY WIND TUNNEL COMPONENTS STRUCTURAL EVALUATION. VOL--ETC(U)
MAY 79 D PETERSON , E WESTERMANN N60921-78-C-0013
UNCLASSIFIED ODAI-1270-8-79-VOL-2 NL

2 OF 3
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PITTSBURGH, PENNSYLVANIA

BY LFP DATE 6/27/78 SUBJECT MACH 14/18 Heater Vessel Bottom End SHEET NO. 1 OF 1
CHKD. BY DATE PROJ. NO 3/1-70

Ref: ϕDANDGA-6/29/78 and ϕDANDAV-6/28/78

Summary of Forces on Main Cylinder Threads

Thread	ΣF_y (Lbs/rad)	
1	356,468	← Max (No. 1)
2	352,199	
3	303,957	
4	265,715	
5	237,652	
6	217,081	
7	200,350	
8	184,894	
9	169,595	
10	154,211	
11	138,863	
12	123,797	
13	109,264	
14	95,480	
15	82,601	
16	70,731.8	
17	59,918.7	
18	50,171.2	
19	41,463.8	
20	33,743.4	
21	26,943.1	
22	20,992.41	$(p = 46{,}000\ psi)$
23	15,825.3	
24	11,410.28	
25	7,836.67	
26	5,709.3	

$$\Sigma F_y(\text{Total}) = 3{,}336{,}872.96 \text{ Lbs/rad}$$

$$[\Sigma F_y(\text{Total})]\cdot\cos(7^\circ) = 3{,}312{,}000.415 \text{ Lbs/rad}$$

$$F_p = \frac{(46{,}000)\pi(24)^2}{4(2\pi)} = 3{,}312{,}000 \text{ Lbs/rad}$$

Agree!

$$\Sigma F_y(AVE) = \frac{\Sigma F_y(\text{Total})}{26} = 128{,}341.2677 \text{ Lbs/rad}$$

$$\frac{\Sigma F_y(\text{Max})}{\Sigma F_y(\text{Ave})} = 2.7775$$

46 1320

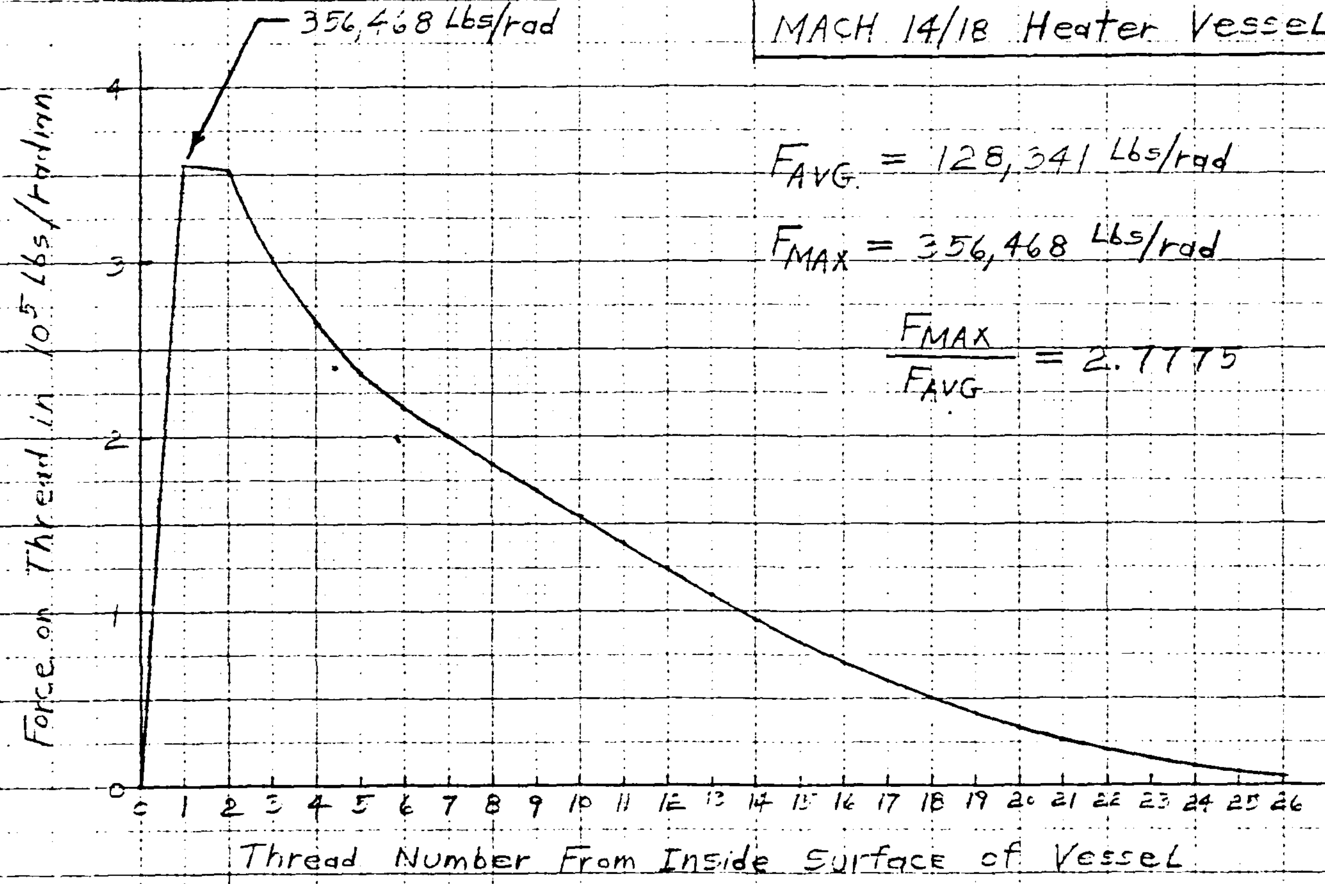
6/-7/78 DEF

Thread Load Distribution
For Bottom End of
MACH 14/18 Heater Vessel

$F_{AVG} = 128,341$ Lbs/rad

$F_{MAX} = 356,468$ Lbs/rad

$$\frac{F_{MAX}}{F_{AVG}} = 2.7775$$



The detailed thread model described in Section 5.2.2 in the main body of this report, which includes the elliptical undercut on the first thread, was used to calculate the maximum stresses in the first thread. The detailed thread model described in Section 5.2.3 in the main body of this report, which has geometry typical of the second and subsequent threads, was used to calculate the maximum stresses in the threads other than the first thread. The results obtained from this evaluation procedure are shown in the following table.

M 14/18 HEATER VESSEL BOTTOM END
Original Design - P = 46,000 psi

Thread No.	Load (lbs/Radian)	Stress Range (psi)
1	356,468.	308,628.*
2	352,199.	378,338.
4	265,715.	281,467.
7	200,350.	202,242.
10	154,211.	144,921.

*Maximum Surface Stress Intensity from Model with Elliptical Undercut

These results indicate that the highest stress occurs in the second thread.

BY DPI DATE 6-9-?? SUBJECT MACH 14/18 Heater Vessel Bottom End SHEET NO 1 OF 1

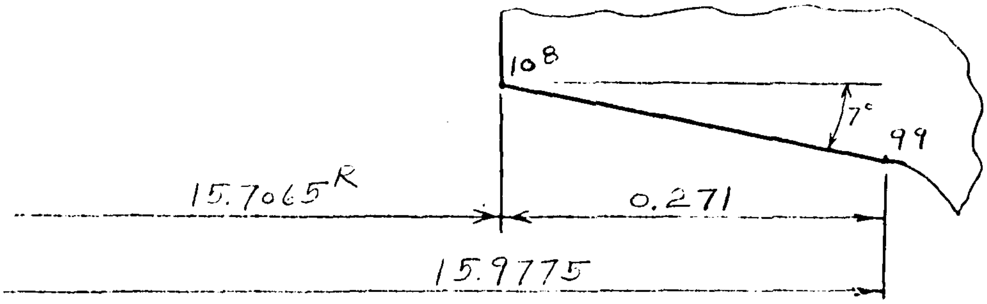
CHKD. BY DATE PROJ. NO JPL-70

Calculate Maximum Equivalent Pressure on Thread

The Maximum Thread load occurs on 1st Thread.

The Total Force on Thread No. 1 (body) from the overall Model = 356,468 Lbs/rad

Total $F = 2\pi(356{,}468)$ Lbs



$$\text{Area} = \frac{\pi\left[(15.9775)^2 - (15.7065)^2\right]}{\cos(7^\circ)}$$

$$P_{Max} = \frac{F}{\text{Area}} = \text{Max. Pressure}$$

$$P_{Max} = \frac{2\cancel{\pi}\,(356{,}468)\cdot\cos(7^\circ)}{\cancel{\pi}\left[(15.9775)^2-(15.7065)^2\right]} = 82{,}412.29 \text{ psi}$$

$$P_{Max} = 82{,}412.29 \text{ psi}$$

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BY DBP DATE SUBJECT M14 Heater Vessel SHEET NO 1 OF 1
CHKD. BY DATE Bottom End PROJ. NO JP1270

Equivalent Thread Pressures for Original Design, Bottom End of Mach 14/18 Heater Vessel

THREAD No.	Thread Load (Lbs/Radian	Thread Pressure (psi)
1	356,468	82,412.29
2	352,199	81,425.33
4	265,715	61,430.99
7	200,350	46,319.17
10	154,211	35,652.24

$$P = \frac{2(\text{THREAD LOAD}) \cdot \cos(7^\circ)}{\left[(15.9775)^2 - (15.7065)^2\right]}$$

$$P = 0.23119259(\text{THREAD LOAD})$$

BY DBP DATE 6/30/78 SUBJECT MACH 14/18 Heater Vessel SHEET NO 1 OF 6
CHKD. BY DATE PROJ. NO JP1270

Determine Material Constant, δ

The stress distribution across a section containing a cir-

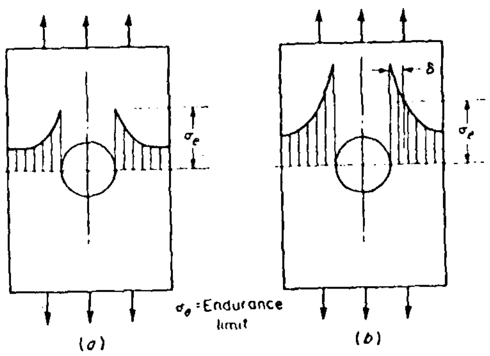


Fig. 6.29. Stress Distribution about a Circular Hole in a Bar

cular hole, Fig. 6.29*a*, has a high stress gradient at the edge of the hole. If the load is just sufficient to bring the peak stress up to the endurance limit, a fatigue failure would hardly be expected since the volume of material at this stress is zero. A finite volume of material must be at the endurance limit before a crack will form, and to obtain this volume of material the endurance limit stress must exist at some finite depth, δ, below the surface; therefore, the steeper the stress gradient, the higher the load required to produce fatigue failure, Fig. 6.29*b*.

The dimension, δ, is a property of the material; and, in general, hard, fine-grained materials have small values of δ, whereas soft, coarse-grained materials have larger values. The relationship between δ and steel tensile strengths, based on correlating fatigue data and the shear theory of failure is shown in Fig. 6.30.

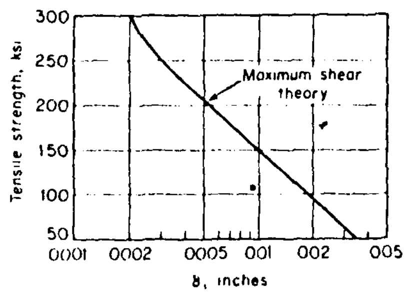


Fig. 6.30. Material Constant δ vs. Tensile Strength for Steel

For the body material, the Tensile strength is 145 Ksi and δ is Equal to 0.00105 inches.

BY L.F.F. DATE 6/9/78 SUBJECT MACH 14/18 Heater Vessel SHEET NO 2 OF 6
CHKD. BY DATE PROJ. NO JPI-70

Calculate Stress Intensity at Depth δ

Ref: Timoshenko and Goodier, Theory of Elasticity, p. 90

The stress distribution in the Vicinity of a small Circular hole in the middle of a plate subjected to Uniform Tension is given by:

$$\sigma_r = S/2\left[1-(a/r)^2\right] + S/2\left[1+3(a/r)^4-4(a/r)^2\right]\cos 2\theta$$

$$\sigma_\theta = S/2\left[1+(a/r)^2\right] - S/2\left[1+3(a/r)^4\right]\cos 2\theta$$

$$\tau_{r\theta} = -S/2\left[1-3(a/r)^4+2(a/r)^2\right]\sin 2\theta$$

when $\theta = 0$, $\tau_{r\theta} = 0$ and the principal stresses are:

$$\sigma_r = S/2\left[2+3(a/r)^4-5(a/r)^2\right]$$

$$\sigma_\theta = S/2\left[-3(a/r)^4+(a/r)^2\right]$$

The stress Intensity is given by:

$$S.I. = |\sigma_r - \sigma_\theta| = S/2\left[2+6(a/r)^4-6(a/r)^2\right]$$

$$= S\left[1+3(a/r)^4-3(a/r)^2\right]$$

Assume that the stress intensity distribution at the thread root radius has the same form as the above stress intensity distribution:

$$S.I. = S\left[1+A(a/r)^4-B(a/r)^2\right]$$

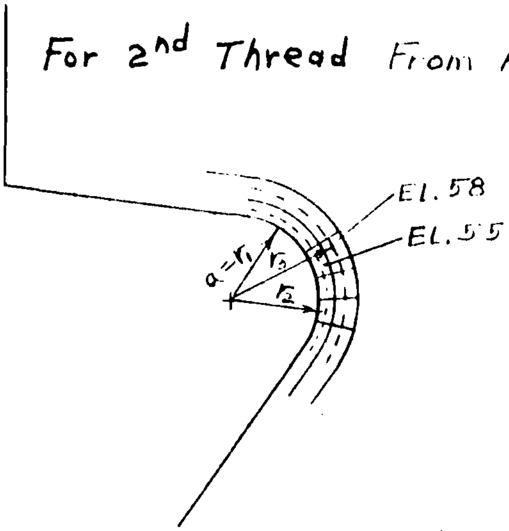
Where: a = Thread Root Radius = 0.108 in.

$r = a + \delta$, in.

δ = Distance from Surface, in.

S, A and B are three unknown Constants.

For 2nd Thread From ANSYS Run ΦDANDYF - 1/12/79



$a = r_1 = 0.108$ in.

$r_2 = 0.118$ in.

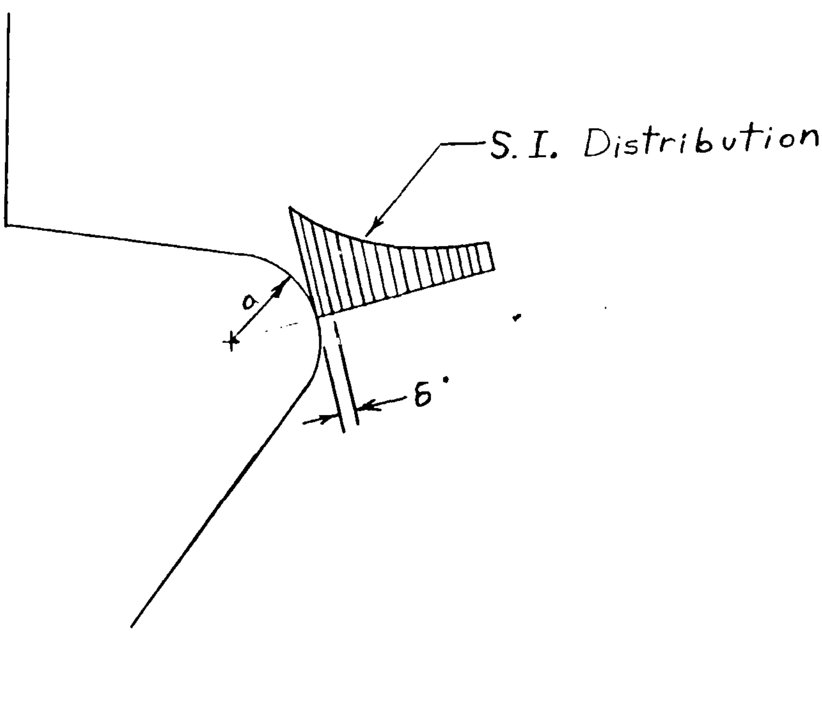
$r_3 = 0.143$ in.

At $(a/r_1) = 1$, S.I. = **191,388 psi**

At $(a/r_2) = 0.91525$, S.I. = **141,227 psi**

At $(a/r_3) = 0.75524$, S.I. = **79,501 psi**

The Known Stress Intensities at the above three Locations can be used to evaluate the three unknowns in the Stress Intensity Distribution Equation.



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BY DBP DATE 5/23/79 SUBJECT MACH. 14/18 Heater Vess SHEET NO 4 OF 6
CHKD. BY DATE PROJ. NO JH 1-70

$$S.I. = S\left[1 + A(a/r)^4 - B(a/r)^2\right]$$

Solving For S, A & B:

$S = 34,664.2913$

$A = 5.2456$

$B = 0.7244$

$$S.I. = 34,664.2913\left[1 + 5.2456(a/r)^4 - 0.7244(a/r)^2\right]$$

At $(a/r) = 1$ S.I. = 191,388 psi

At $(a/r) = 0.91525$, S.I. = 141,225 psi

At $(a/r) = 0.75524$, S.I. = 79,499.9 psi

At $r = a + \delta$

$$r = 0.108 + 0.00105 = 0.10905 \text{ in.}$$

$$\frac{a}{r} = \frac{0.108}{0.10905} = 0.99037$$

$$S.I. = 34{,}664.2913\left[1 + 5.2456\,(0.99037)^4 - 0.7244\,(0.99037)^2\right]$$

$$S.I. = 184{,}965 \text{ psi}$$

This stress intensity must be multiplied by the following Factor to Account for the interrupted Threads on the Bottom End:

$$\text{Factor} = \left(\frac{90}{44}\right) = 2.0455 \quad \left\{\begin{array}{l}\text{Due to } 44^\circ \text{ interrupted} \\ \text{Thread in every } 90^\circ \text{ Arc}\end{array}\right]$$

Therefore, the Stress Intensity at the root of Thread No. 1 on the <u>Bottom End</u> of the body where the Thread Load is a Maximum and Equal to 356,468 lbs/rad is:

$$S.I.(Max) = \left(\frac{90}{44}\right)(184{,}965) = 378{,}338 \text{ psi}$$

Fatigue Life of Threads on Bottom End Closure

$S_{Range}(Max) = 378,338$ psi

$S_{alt} = 189,169$ psi $\qquad S_y = 130,000$ psi

$S'_{mean} = 189,169$ psi $\qquad S_u = 145,000$ psi

Since $S_{alt} > S_y$, $S_{mean} = 0$

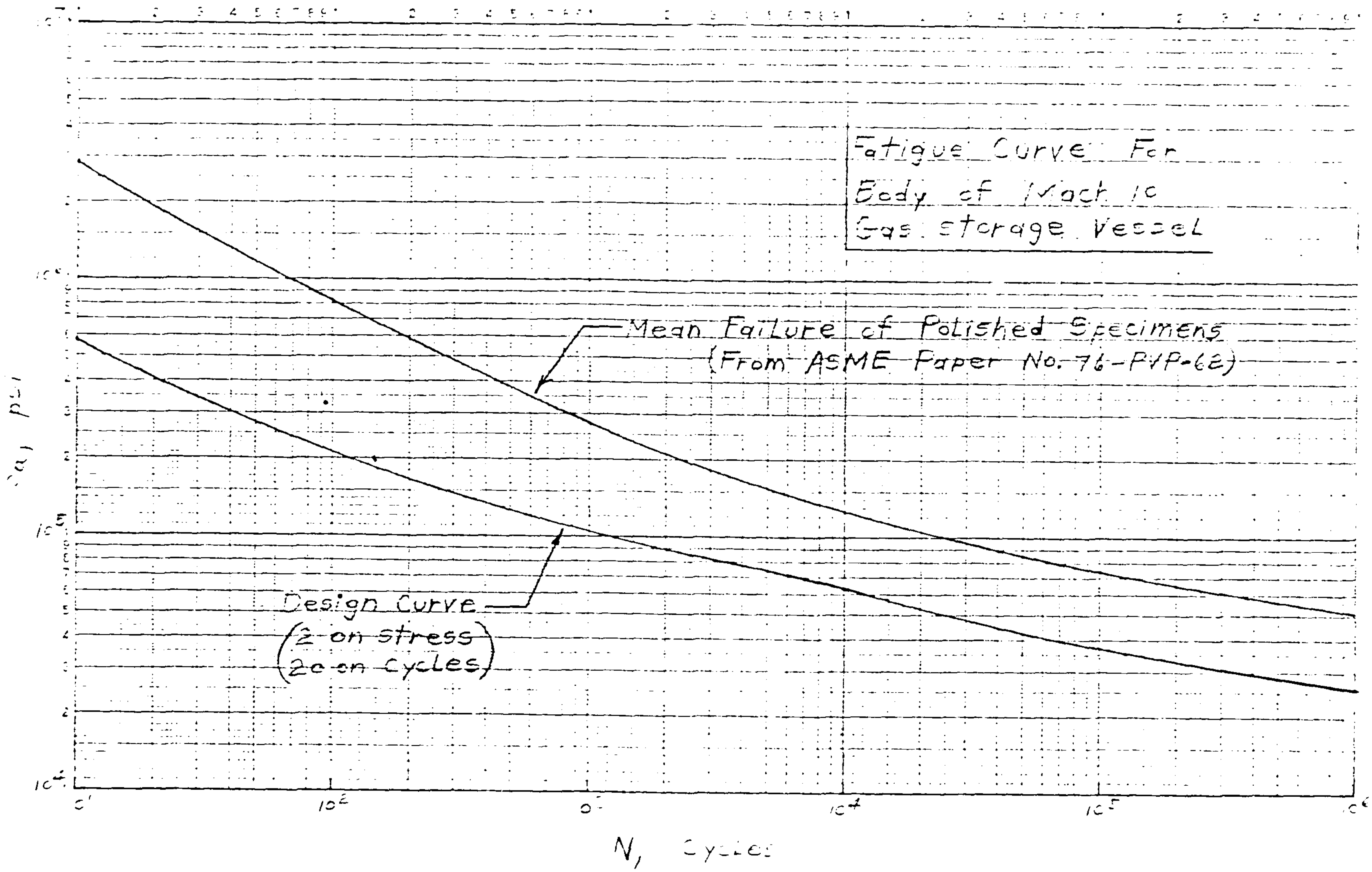
$$S_{eq} = \frac{7 S_{alt}}{8 - \left[1 + \frac{S_{mean}}{S_u}\right]^3} = 189,169 \text{ psi}$$

The Design Life from the Fatigue Data from ASME Paper No. 76-PVP-62 For the body Material with a Factor of 2 on Stress and a Factor of 20 on cycles is:

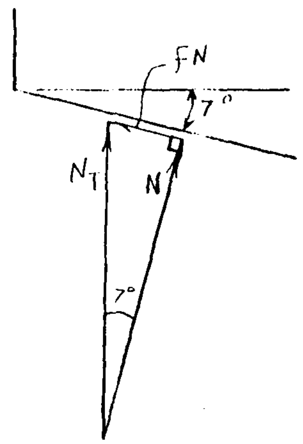
$N = 136$ cycles (For 2nd Thread)

Since the theoretical fatigue curves from this paper were obtained on small polished specimens tested in air, factors must be applied to account for size effects, surface finish, environmental effects, and scatter of data. Therefore, a factor of either 2 on stress or 20 on cycles, whichever is more conservative at each point, was applied to the mean failure curve to obtain a design curve which accounts for these effects. These factors have been confirmed by several fatigue tests and simulated service tests on models of components.

Fatigue Curve For M14/18 Heater Vessel ($S_u = 145$ KSI)



Friction Loading - 2nd Thread - Original Design



Assume $f = \tan(7°)$

$f = 0.122785$

$N_T = (352,199.)(\cos 7°) = 349,573.7621$ Lbs/Radian

$N = (349,573.7621)\cdot(\cos 7°) = 346,968.0923$ Lbs/Radian

$fN = 42,602.325$ Lbs/Radian $(f = \tan 7°)$

Apply FX = -C At Nodes 100 to 107 {8 Nodes)

$$C = \frac{42,602.325}{8} = 5,325.291 \text{ Lbs/Radian}$$

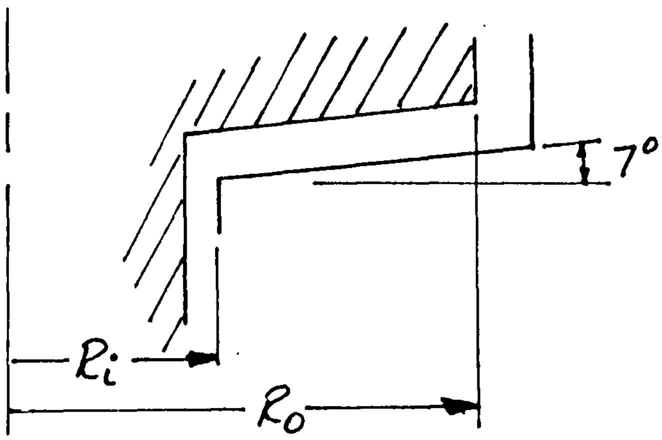
$P_{Max} = 0.231191259\,N = 80,215.99$ psi

The fatigue evaluation of the second thread on the bottom end of the MACH 14/18 Heater Vessel original design was redone for an internal pressure of 28,000 psi for no friction between the threads and for a coefficient of friction equal to 0.122785 (Tan 7°) between the threads. The resulting fatigue design lives for these two cases are shown in the following table.

Location	Stress Range, psi	Calculated Fatigue Design Life, cycles
2nd Thread (No friction)	230,293	575
2nd Thread (With friction)	257,071	455

OUTLET END OF HEATER VESSEL

The following figures show the distribution of forces along the 32"-1 and 25"-1 thread interfaces. In both cases, the maximum load occurs at the first tooth. This load must be converted into an equivalent pressure for use in the detailed tooth model.



Given the total load on the tooth F (lbs/rad), the equivalent pressure (P_{EQ}) is:

$$\frac{\text{AREA}}{\text{RADIAN}} = \frac{\pi(R_o^2 - R_i^2)}{2\pi \cos 7°}$$

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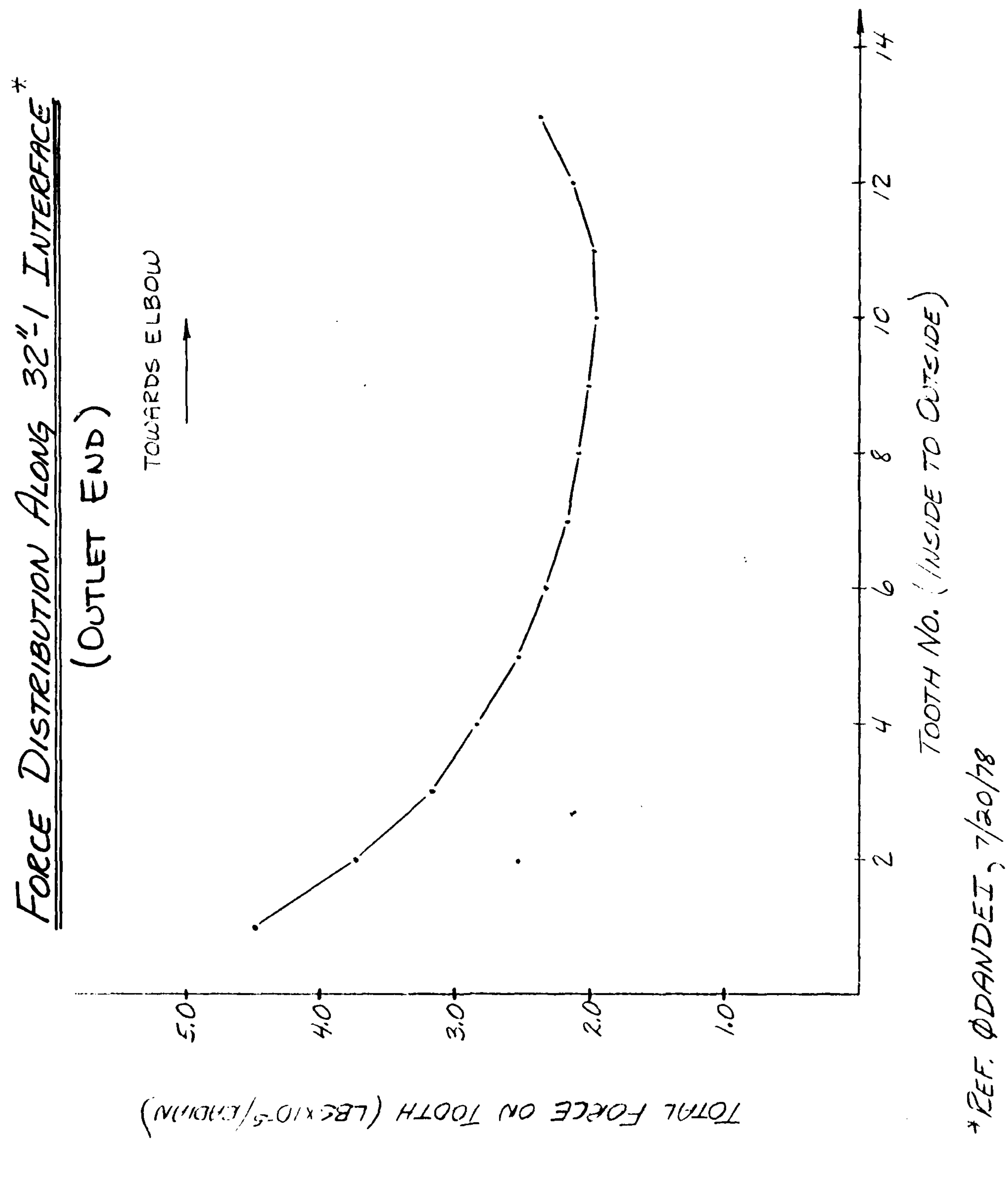
BY ELW DATE 7/20/78 SUBJECT NAVY MACH 15 & 18 HEATER SHEET NO OF

CHKD. BY DATE PROJ. NO J1209



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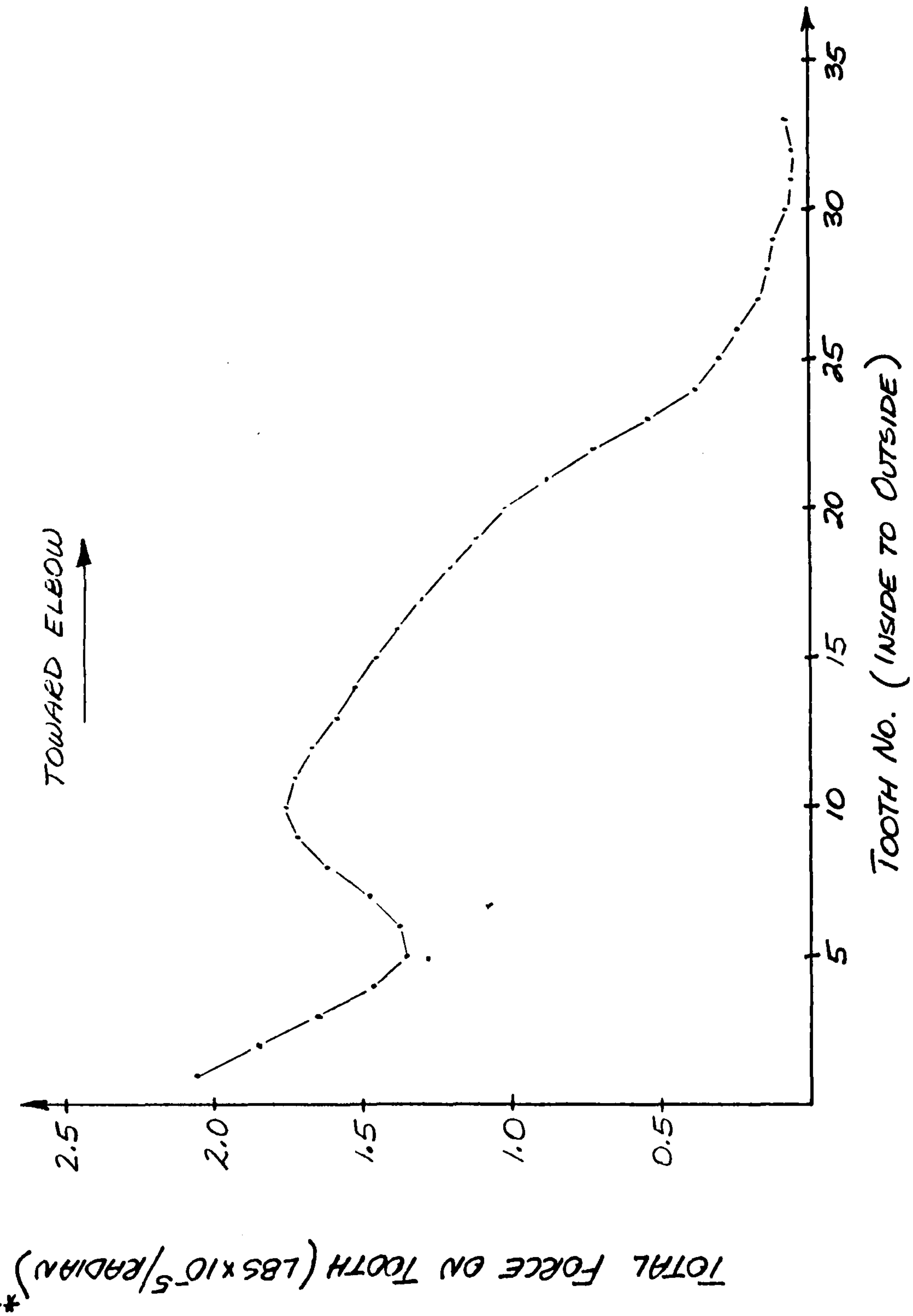
BY ELW DATE 7/20/78 SUBJECT NAVY MACH 15 & 18 HEATER — SHEET NO ___ OF ___
CHKD. BY ___ DATE ___ — PROJ. NO J1209

FORCE DISTRIBUTION ALONG 25"-1 INTERFACE*
(OUTLET END)



*REF. ODANDEI, 7/20/78 **DOES NOT ACCOUNT FOR INTERRUPTED THREADS.

$$\therefore \; P_{EQ} = \frac{2F \cos 7°}{(R_o^{\,2} - R_i^{\,2})}$$

For the 32"-1 threads:

$F = 4.493 \times 10^5$ lbs/rad

$R_i = 15.667$ in.; $R_o = 15.964$ in.[2]

$\therefore \; P_{EQ} = 94{,}949$ psi

For the 25"-1 threads:

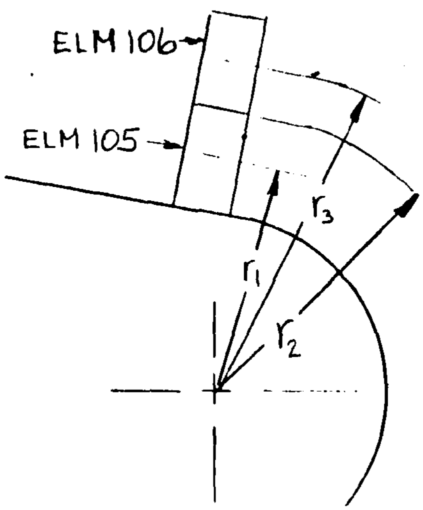
$F = 2.059 \times 10^5$ lbs/rad

$R_i = 12.167$ in.; $R_o = 12.464$ in.

$\therefore \; P_{EQ} = 55{,}872$ psi

The material constant δ is the same as for the bottom end of the heater, $\delta = 0.00105$ inches. The same procedure is followed in determining the stress intensity at the depth δ, as was described for the bottom end.

<u>32"-1 Threads</u>



From ANSYS Run ØDANDRB, 7/24/78

r_1 = .12881

r_2 = .14871

r_3 = .16861

At (a/r_1) = .826, σ_I = 195,073 psi

(a/r_2) = .730, σ_I = 148,668 psi

(a/r_3) = .634, σ_I = 85,024 psi

Therefore, the assumed stress distribution in the vicinity of the thermal root radius is:

$$\sigma_I = S[1 + A(\frac{a}{r})^4 - E(\frac{a}{r})^2]$$

where a = thermal root radius = 0.10891 in.

r = a + δ

δ = distance from surface, in.

S,A,E = constants to be determined

Using the above three equations and solving for S,A,E yields:

S = 244,357

A = 2.560

E = 4,382

Therefore, for the fatigue analysis, the maximum stress intensity is:

$$\sigma_I = 204{,}127 \text{ psi}$$

The stress intensity range for one pressure cycle is:

$$\sigma_{RANGE} = 204{,}127 \text{ psi}, \quad \sigma_y = 130{,}000 \text{ psi}$$

$$\sigma_{ALT} = 102{,}063 \text{ psi}, \quad \sigma_u = 145{,}000 \text{ psi}$$

$$\sigma_{MEAN} = 102{,}063 \text{ psi}$$

Snow, A. L. and Langer, B. F., "Low Cycle Fatigue of Large Diameter Bolts," ASME J. of Engrg. for Industry, Feb. 1967.

$$\sigma_{ALT} + \sigma_{MEAN} = 204{,}127 \text{ psi}$$

Since $\sigma_{ALT} < \sigma_y$ and $\sigma_{ALT} + \sigma_{MEAN} > \sigma_y$,

$$\sigma_{MEAN} = \sigma_y - \sigma_{ALT} = 130{,}000 - 102{,}063 = 27{,}936 \text{ psi}$$

$$\sigma_{eq} = \frac{7\sigma_{ALT}}{8 - \left[1 + \frac{\sigma_{MEAN}}{\sigma_u}\right]^3} = \frac{(7)(102{,}063)}{8 - \left[1 + \left(\frac{27{,}936}{145{,}000}\right)\right]^3}$$

$$\sigma_{eq} = 113{,}340 \text{ psi}$$

This equivalent stress will be used to enter the fatigue curve. This curve is from ASME Paper No. 76-PVP-62. Since the theoretical fatigue curves from this paper were obtained on small polished specimens tested in air, factors must be applied to account for size effects and scatter in the date. Therefore, a factor of either 2 on stress or 20 on cycles, whichever is more conservative at each point, was applied to the mean failure

curve to obtain a design curve which accounts for these effects. The Design Life for a σ_{eq} of 113,340 psi is:

$$N = 650 \text{ cycles}$$

<u>25"-1 Threads</u>

Following the same procedure as outlined in the previous section for the 32"-1 threads, the maximum stress intensity is: (Ref ANSYS Run ODAND5Z, 7/24/78)

$$\begin{aligned} \text{At}\quad (a/r_1) &= .826,\ \sigma_I = 130{,}986 \text{ psi} \\ (a/r_2) &= .730,\ \sigma_I = 105{,}039 \\ (a/r_3) &= .634,\ \sigma_I = 94{,}835 \end{aligned}$$

$$\sigma_I = 228{,}607 \text{ psi}$$

The 25"-1 threads are interrupted, therefore, this stress value must be increased by

$$\frac{90}{44} = 2.045$$

since the finite element model assumed the threads to be continuous. Therefore:

$$\sigma_I = 467{,}501 \text{ psi}$$

The stress range is σ_{RANGE} = 467,501 psi

σ_{ALT} = 233,750 psi; σ_y = 130,000 psi

σ_{MEAN} = 233,750 psi; σ_u = 145,000 psi

$\sigma_{ALT} + \sigma_{MEAN}$ = 467,501

Since $\sigma_{ALT} > \sigma_y$,

$\sigma_{eq} = \sigma_{ALT}$ = 233,750 psi

This equivalent stress will be used to enter the fatigue curve. This curve is from ASME Paper No. 76-PVP-62. Since the theoretical fatigue curves from this paper were obtained on small polished specimens tested in air, factors must be applied to account for size effects and scatter in the date. Therefore, a factor of either 2 on stress or 20 on cycles, whichever is more conservative at each point, was applied to the mean failure curve to obtain a design curve which accounts for these effects. The Design Life for a σ_{eq} of 233,750 psi is:

N = 70 cycles

APPENDIX 4C

FRACTURE MECHANICS EVALUATION
OF THREADS
for
MACH 14/18 HEATER VESSEL
ORIGINAL DESIGN

Fracture Mechanics Evaluation

The procedure followed herein is outlined in detail in Appendix 5C.

The thread material is modified AISI 4340, or "gun steel." This is now designated ASTM A-723 material. Assume this material has the following properties:

$$S_u = 145{,}000 \text{ psi}$$

$$S_y = 130{,}000 \text{ psi}$$

$$K_{IC} = 100 \text{ Ksi}\sqrt{\text{in.}}$$

The calculation of the critical crack sizes and the curves of cycles to failure for various initial defect sizes for the threads on the top and bottom ends are given on the following pages.

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BY DBP DATE 5/23/79 SUBJECT MACH 14/18 Heater Vessel Bottom End SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP1270

Threads on Bottom End Closure

For 2nd Thread for $P = 46{,}000$ psi

$\sigma = \Delta\sigma = 378{,}338$ psi , $K_{IC} = 100\ ksi\sqrt{in}$

1. $K_{IC} = 100\ ksi\sqrt{in}$

2. Critical Crack Depth

$$a_{cr} = \frac{1}{1.25\pi}\left(\frac{100{,}000}{378{,}338}\right)^2 = 0.017790''$$

3. Cycles to Failure

$C_o = 1.1766441 \times 10^{-15}$ for ΔK in $psi\sqrt{in}$

$(n-2) = 0.25 \qquad M^{n/2} = (1.25\pi)^{1.125} = 4.659264564$

$\Delta\sigma^n = (378{,}338)^{2.25} = 3.550012795 \times 10^{12}$

$$\frac{1}{a_{cr}^{(n-2)/2}} = \frac{1}{(0.01779)^{0.125}} = 1.654733299$$

$$N = 412.097067\left[\frac{1}{a_i^{0.125}} - 1.654733299\right]$$

$$a_i = \left(\frac{412.097067}{N + 681.910060}\right)^8$$

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BY DBP DATE 5/23/79 SUBJECT Mach 14/18 Heater Vessel Bottom End SHEET NO 2 OF 2
CHKD. BY DATE PROJ. NO JP1270

a_i Versus N for Threads
on Bottom End Closure
$\sigma = \Delta\sigma = 378{,}338$ psi, $K_{IC} = 100$ Ksi$\sqrt{in}$
Modified AISI 4340 Material

a_i inches	N cycles
0.0158343	10
0.0141171	20
0.0101003	50
0.00814084	70
0.00595308	100
0.00227301	200
0.00021843	500
0.00006254	700
0.0000129889	1000
0.0000016192	1500
0.000000310773	2000

$$a_i = \left(\frac{412.097067}{N + 681.910060}\right)^8$$

46 5490

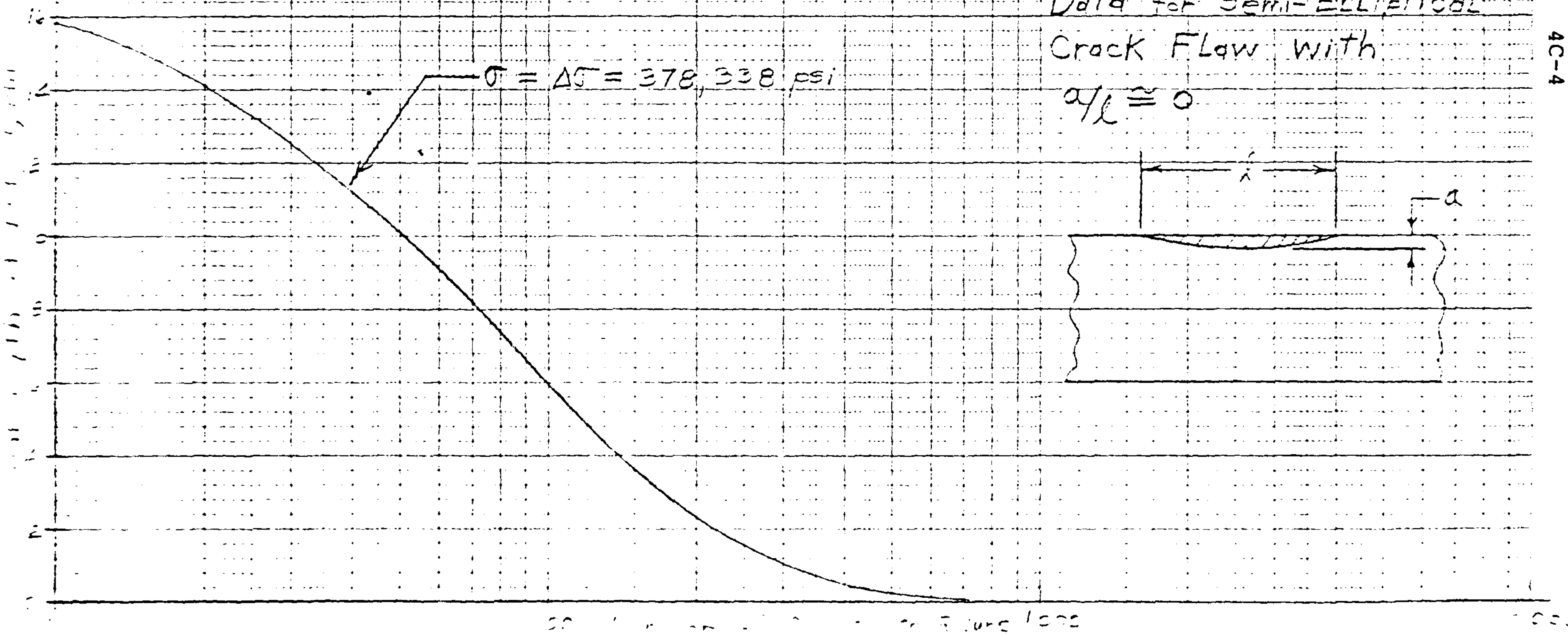
Fracture Mechanics Evaluation of Threads on Bottom End Closure of MACH 14/18 Heater Vessel

$K_{IC} = 100\ Ksi\sqrt{in}$

$\frac{da}{dN} = 1.17366 \times 10^{-15} (\Delta K)^{2.25}$

Data for Semi-Elliptical Crack Flaw with

$a/\ell \cong 0$



25"-1 Threads on Outlet End Closure

If $\sigma = \Delta\sigma = 467{,}500$ psi and $K_{IC} = 100$ Ksi$\sqrt{\text{in.}}$

1. $K_{IC} = 100$ Ksi$\sqrt{\text{in.}}$

2. Critical Crack Depth

$$a_{CR} = \frac{1}{1.25\pi}\left(\frac{100{,}000}{467{,}500}\right)^2 = 0.0116 \text{ in.}$$

3. Cycles to Failure

$$C_o = 1.1736 \times 10^{-15} \text{ for } \Delta K \text{ in psi}\sqrt{\text{in.}}$$

$$(n-2) = 0.25, M^{N/2} = (1.25\pi)^{1.125} = 4.6593$$

$$\Delta\sigma^n = (467{,}500)^{2.25} = 5.715 \times 10^{12}$$

$$\frac{1}{a_{CR}^{(n-2)/2}} = \frac{1}{(.0116)^{.125}} = 1.745$$

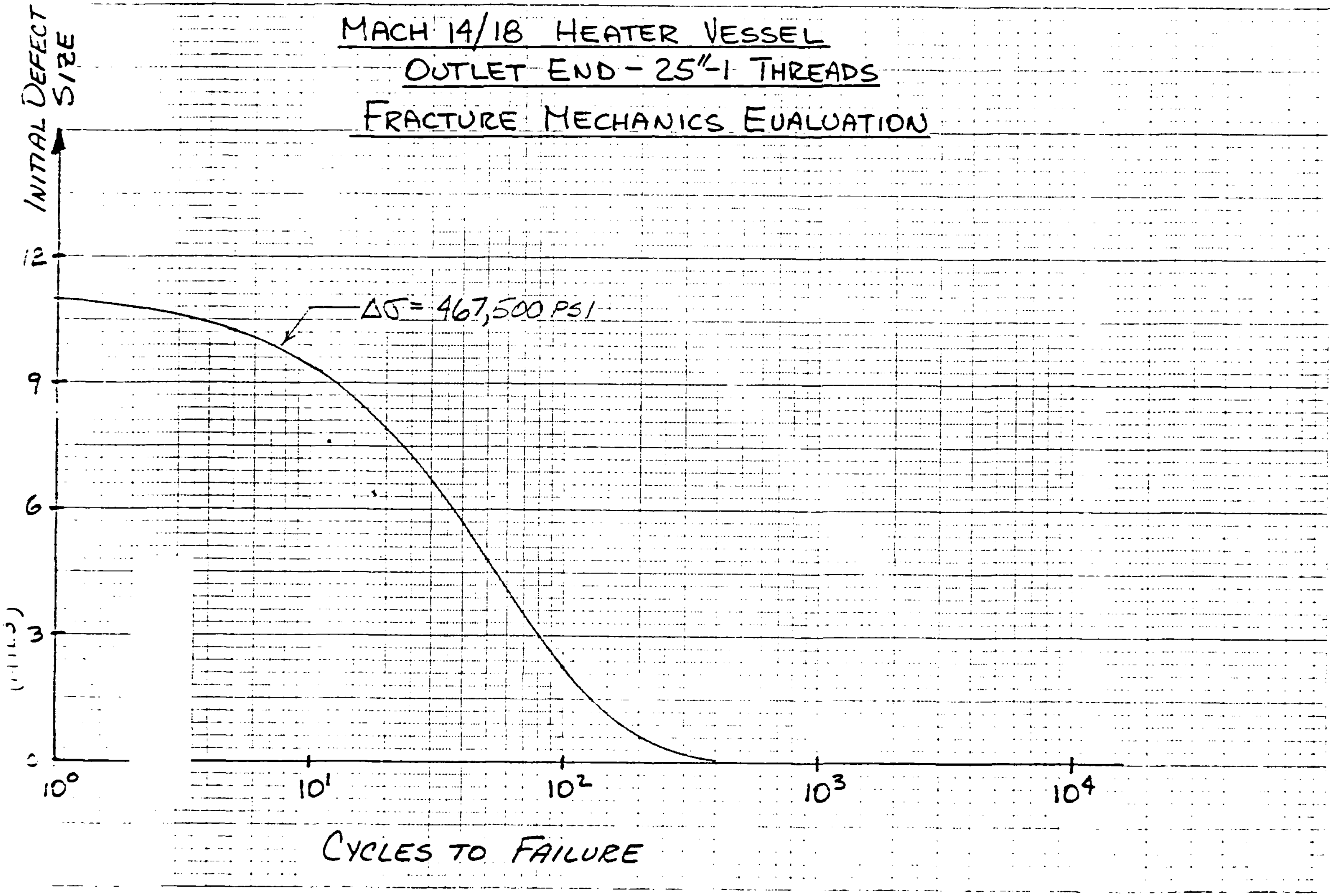
$$N = 255\left[\frac{1}{a_i^{0.125}} - 1.745\right]$$

$$a_i = \left[\frac{255}{N + 447}\right]^8$$

A_i vs. N for 25"-1 Threads on Outlet End Closure

$\sigma = \Delta\sigma = 467{,}500$ psi, $K_{IC} = 100$ KSI$\sqrt{\text{in.}}$

a_i Inches	N Cycles
.01102	1
.01082	2
.01026	5
.00939	10
.00223	100
2.764×10^{-5}	500
9.302×10^{-7}	1,000
2.307×10^{-11}	5,000
1.260×10^{-13}	10,000


MACH 14/18 HEATER VESSEL
OUTLET END - 25"-1 THREADS
FRACTURE MECHANICS EVALUATION
INITIAL DEFECT SIZE
12
9
6
3
0
$\Delta\sigma$ = 467,500 PSI
10^0
10^1
10^2
10^3
10^4
CYCLES TO FAILURE

APPENDIX 5A

PRIMARY STRESS EVALUATION

for

DRIVER VESSEL

1. Primary Stresses in Cylinder and Liner

The primary stresses in the cylinder and liner section of the Gas Storage Vessel due to an internal pressure of 60,000 psi and a shrink fit of 0.021" on the radius between the liner and the cylinder were calculated using a special-purpose computer program. The resulting stresses are listed and compared to the allowable stresses on the following page.

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BY LBF DATE 2/?/78 SUBJECT Gas Storage Vessel SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Ref: ΦDANDGP - 1/9/78 (ELW)

Liner Stresses Compared
to the Allowable Stresses

Stress Category	Calculated Stress (psi)	Allowable Stress (psi)
P_m	94,677	$S_m = 80,000$
$P_m + P_b$	132,750	$1.5S_m = 120,000$

Stresses in Liner Are Due to Internal Pressure of 60,000 psi and Shrink Fit of 0.021" on Radius

S_u = 160,000 psi for Liner

$$S_m = \frac{S_u}{2} = 80,000 \text{ psi}$$

Cylinder Body Stresses Compared
to the Allowable Stresses

Stress Category	Calculated Stress (psi)	Allowable Stress (psi)
P_m	73,604	$S_m = 72,500$
$P_m + P_b$	100,363	$1.5S_m = 108,700$

Stresses in Cylinder Body Are Due to Internal Pressure of 60,000 psi and Shrink Fit of 0.021" on Radius

S_u = 145,000 psi for Cylinder Body

$$S_m = \frac{S_u}{2} = 72,500 \text{ psi}$$

2. Maximum Stresses in Liner and Cylinder Body

The maximum stress intensities in the liner and cylinder body due to an internal pressure of 60,000 psi and a shrink fit of 0.021" on the radius between the liner and cylinder body were calculated by hand. These hand calculations are given on the following pages. The resulting stresses are summarized in two tables at the end of this section.

BY Lt I DATE 1-1-78 SUBJECT Gas Storage Vessel SHEET NO 1 OF 4
CHKD. BY DATE PROJ. NO JH270

Maximum Stresses in Liner And Cylinder Body

Reference: Strength of Materials, Part II, Timoshenko, pp. 211-214.

$$\sigma_t = \frac{a^2 p_i}{b^2 - a^2}\left(1 + \frac{b^2}{r^2}\right) \quad \{\text{Tangential or Hoop stress}\}$$

$$\sigma_r = \frac{a^2 p_i}{b^2 - a^2}\left(1 - \frac{b^2}{r^2}\right) \quad \{\text{Radial stress}\}$$

1. Pressure Stresses

$a = 12''$ $b = 24''$ $p_i = 60{,}000$ psi

(a) At Inside Surface of Liner ($r = 12''$)

$$\sigma_t = \frac{(12)^2(60{,}000)}{(24)^2 - (12)^2}\left[1 + \left(\frac{24}{12}\right)^2\right]$$

$$\sigma_t = 20{,}000\left[1 + \left(\frac{24}{12}\right)^2\right] = 100{,}000 \text{ psi}$$

$$\sigma_r = -p_i = -60{,}000 \text{ psi}$$

$$S = \sigma_t - \sigma_r = 160{,}000 \text{ psi} \quad \{\text{stress Intensity}\}$$

(b) At Inside Surface of Cylinder ($r = 17.5''$)

$$\sigma_t = 20{,}000\left[1 + \left(\frac{24}{17.5}\right)^2\right] = 57{,}616 \text{ psi}$$

$$\sigma_r = 20{,}000\left[1 - \left(\frac{24}{17.5}\right)^2\right] = -17{,}616 \text{ psi}$$

$$S = \sigma_t - \sigma_r = 75{,}232 \text{ psi}$$

Maximum Stresses in Liner And Cylinder Body (continued)

2. Shrink Fit Stresses

$a = 12''$ $b = 17.5''$ $c = 24''$

$\delta = 0.021''$ $E = 30 \times 10^6$ psi

(a) Shrink Fit Pressure

$$p = \frac{E\delta(b^2-a^2)(c^2-b^2)}{2b^3(c^2-a^2)}$$

$$p = \frac{(30\times10^6)(0.021)\left[(17.5)^2-(12)^2\right]\left[(24)^2-(17.5)^2\right]}{2(17.5)^3\left[(24)^2-(12)^2\right]}$$

$$p = 5{,}955 \text{ psi}$$

(b) At Inside Surface of Liner ($r = 12''$)

$$\sigma_t = -\frac{2pb^2}{b^2-a^2} = -\frac{2(5{,}955)(17.5)^2}{(17.5)^2-(12)^2} = -22{,}480 \text{ psi}$$

$$\sigma_r = 0$$

(c) At Inside Surface of Cylinder ($r = 17.5''$)

$$\sigma_t = \frac{p(b^2+c^2)}{c^2-b^2} = \frac{(5{,}955)\left[(17.5)^2+(24)^2\right]}{(24)^2-(17.5)^2}$$

$$\sigma_t = 19{,}477 \text{ psi}$$

$$\sigma_r = -p = -5{,}955 \text{ psi}$$

Maximum Stresses in Liner And Cylinder Body (continued)

3. Pressure Plus Shrink Fit Stresses

(a) At Inside Surface of Liner ($r = 12''$)

$$\sigma_t = 100,000 - 22,480 = 77,520 \text{ psi}$$

$$\sigma_r = -60,000 + 0 = -60,000 \text{ psi}$$

$$S = \sigma_t - \sigma_r = 137,520 \text{ psi}$$

(b) At Inside Surface of Cylinder ($r = 17.5''$)

$$\sigma_t = 57,616 + 19,477 = 77,093 \text{ psi}$$

$$\sigma_r = -17,616 - 5,955 = -23,571 \text{ psi}$$

$$S = \sigma_t - \sigma_r = 100,664 \text{ psi}$$

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BY DBF DATE 2/2/78 SUBJECT Gas storage Vessel SHEET NO 4 OF 4
CHKD. BY DATE PROJ. NO JPL-70

Stresses in Liner and Cylinder Body Due to 60,000 psi Internal Pressure only

	At Inner Surface of Liner	At Inner Surface of Cylinder
σ_t Hoop Stress (psi)	100,000	57,616
σ_r Radial Stress (psi)	-60,000	-17,616
S Stress Intensity (psi)	160,000	75,232

Stresses in Liner and Cylinder Body Due to 60,000 psi Internal Pressure Plus 0.021" Shrink Fit

	At Inner Surface of Liner	At Inner Surface of Cylinder
σ_t Hoop Stress (psi)	77,520	77,093
σ_r Radial Stress (psi)	-60,000	-23,571
S Stress Intensity (psi)	137,520	100,664

APPENDIX 5B

FATIGUE EVALUATION OF THREADS

for

DRIVER VESSEL

ORIGINAL DESIGN

FATIGUE ANALYSIS OF THREADS

The fatigue analysis calculations used to calculate the fatigue design life of the threads are given on the following pages. The calculations are divided into the following parts:

(a) Summary of Loads on Main Cylinder Threads on Outlet and Inlet Ends

(b) Equivalent Pressure Calculation for Maximum Thread Load on Outlet End

(c) Fatigue Analysis of Stress Gradient at Thread Root Radius of 2nd Thread on Outlet End

(d) Fatigue Life of Threads on Outlet End Closure

(e) Stress Results for Threads 1, 2, 8, and 9 on Inlet End

(f) Fatigue Life of Threads on Inlet End Closure

(g) Fatigue Curve for Body Material of the Driver Vessel

(h) Summary of Fatigue Design Lives for Inlet and Outlet Ends

At P = 60,000 psi with no friction, a fatigue design life of 680 cycles was obtained for the threads on the outlet end closure, and a fatigue design life of 133 cycles was obtained for the threads on the inlet end closure.

BY DBP DATE 11/9/78 SUBJECT Gas Storage Vessel Outlet End SHEET NO 1 OF 2
CHKD BY DATE PROJ. NO JP1270

Summary of Forces on Main Cylinder Threads

Thread	ΣF_y (Lbs/rad) X 10^5
1	4.3641
2	4.48381
3	3.97125
4	3.60678
5	3.35787
6	3.16850
7	2.99931
8	2.82762
9	2.64333
10	2.44365
11	2.22929
12	2.00180
13	1.76163
14	1.50643
15	1.23011
16	0.928985

← Max. (No. 2)

$$\Sigma F_y(\text{Total}) = 43.524465 \times 10^5 \text{ Lbs/Radian}$$

$$[\Sigma F_y(\text{Total})] \cdot \cos(7°) = 43.20004 \times 10^5 \text{ Lbs/radian}$$

Agree!

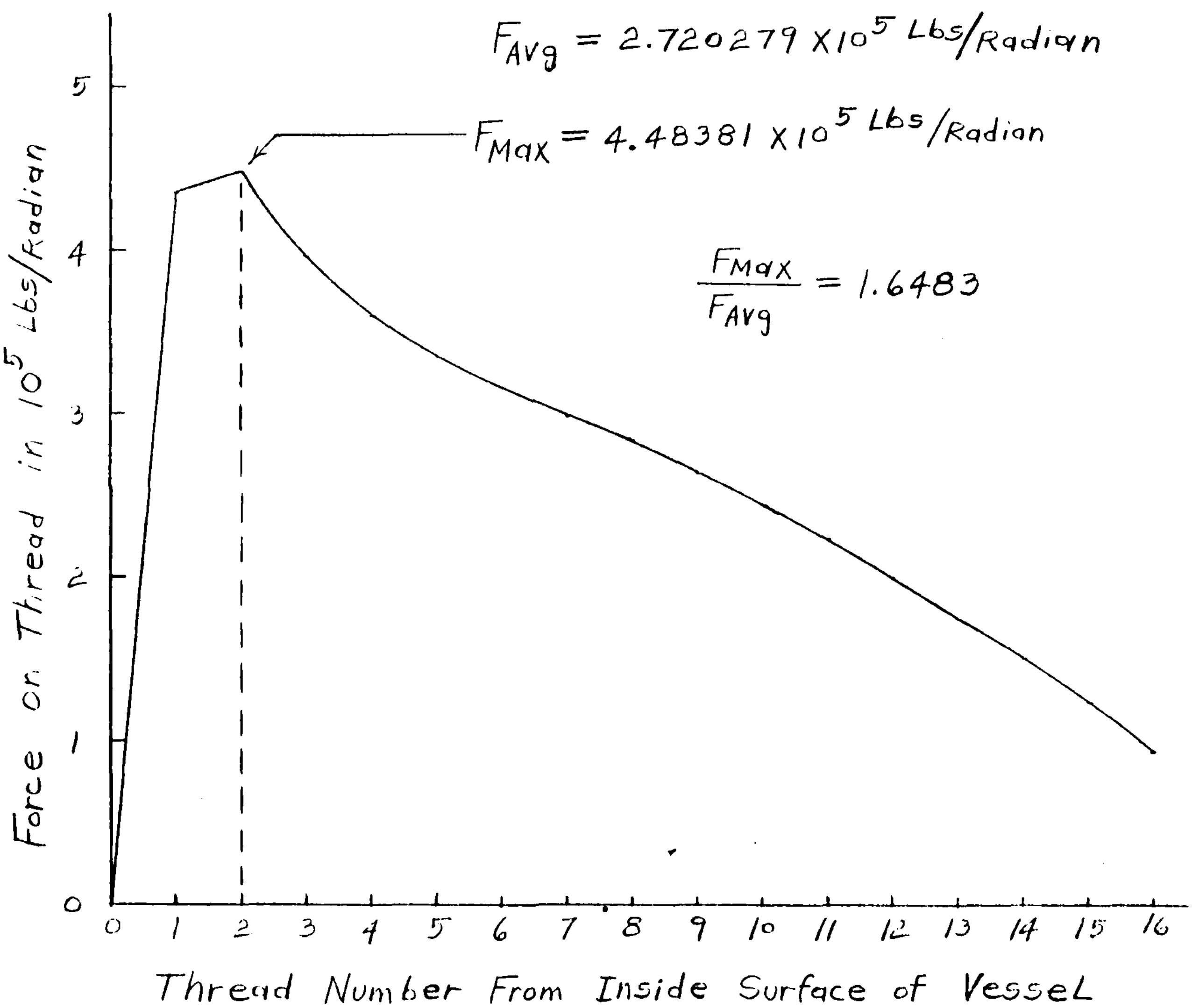
$$p = 60{,}000 \text{ psi}$$

$$F_p = \frac{(60{,}000)\,\pi\,(24)^2}{4\,(2\pi)} = 43.20000 \times 10^5 \text{ Lbs/radian}$$

$$\Sigma F_y(\text{AVE}) = \frac{\Sigma F_y(\text{Total})}{16} = 2.720279063 \times 10^5 \text{ Lbs/Radian}$$

$$\frac{\Sigma F_y(\text{Max})}{\Sigma F_y(\text{Ave})} = 1.6483$$

Gas Storage Vessel
Outlet End
Thread Load Distribution



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BY DBP DATE 11/13/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 1 OF 2
CHKD BY DATE PROJ. NO JP1270

ORIGINAL DESIGN

Summary of Forces on Main Cylinder Threads

Thread	$\sum F_y$ (Lbs/Radian)
1	378,073.
2	390,925. ← Max (No. 2)
3	345,699.
4	313,559.
5	291,436.
6	274,244.
7	259,016.
8	243,857.
9	228,027.
10	211,412.
11	194,212.
12	176,756.
13	159,393.
14	142,437.
15	126,156.
16	110,753.
17	96,369.
18	83,094.
19	70,971.5
20	60,000.2
21	50,153.7
22	41,380.
23	33,609.6
24	26,759.4
25	20,738.2
26	15,446.31
27	10,775.63
28	6,609.54
29	2,823.30
30	-705.6
31	-4,085.187
32	-7,356.38

$$\sum F_y (\text{Total}) = 4,352,538.243 \text{ Lbs/Radian}$$

$$\left[\sum F_y (\text{Total})\right] \cdot \cos(7^\circ) = 43.2 \times 10^5 \text{ Lbs/Radian}$$

Agree!

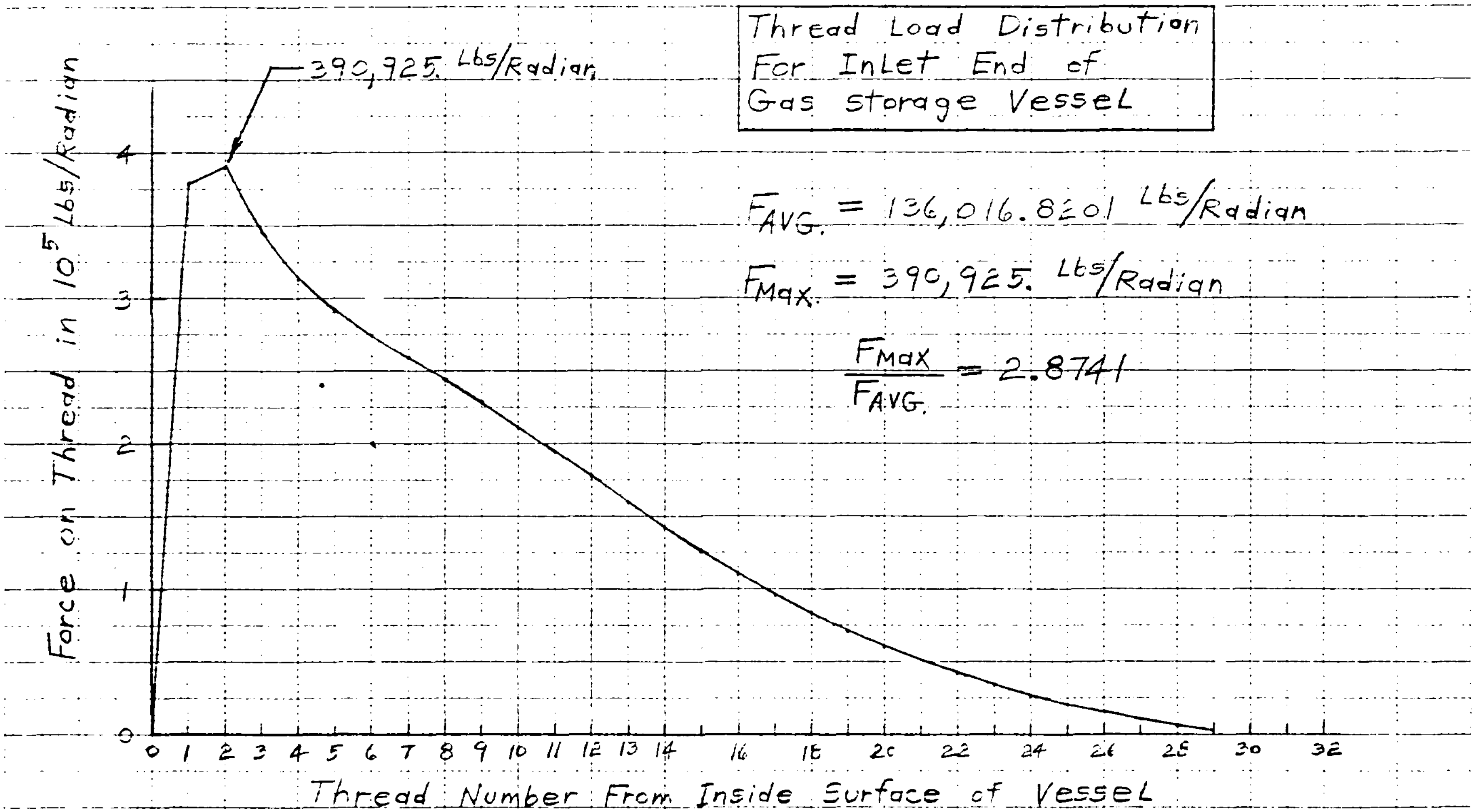
$p = 60,000$ psi

$$F_p = \frac{(60,000)\pi(24)^2}{4(2\pi)} = 43.2 \times 10^5 \text{ Lbs/Radian}$$

Forces on Main Cylinder Threads (continued)

$$\Sigma F_y(AVE) = \frac{\Sigma F_y\,(Total)}{32} = 136{,}016.8201$$

$$\frac{\Sigma F_y\,(Max)}{\Sigma F_y\,(Ave)} = 2.8741$$


Thread Load Distribution
For Inlet End of
Gas storage Vessel
390,925. Lbs/Radian
$F_{AVG.} = 136,016.8201$ Lbs/Radian
$F_{Max.} = 390,925.$ Lbs/Radian
$\frac{F_{Max}}{F_{AVG.}} = 2.8741$
Force on Thread in 10^5 Lbs/Radian
0
1
2
3
4
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 16 18 20 22 24 26 28 30 32
Thread Number From Inside Surface of Vessel

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BY DBP DATE 11/9/78 SUBJECT Gas storage Vessel Outlet End SHEET NO 1 OF 1
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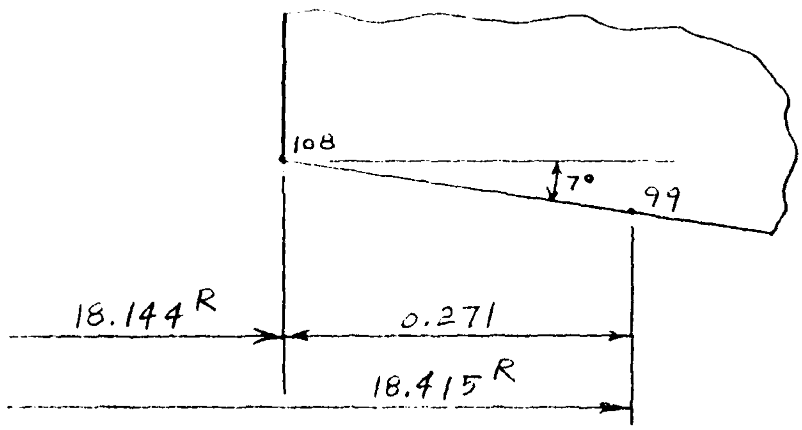
Maximum Equivalent Pressure on 2nd Thread

The Force on Thread No. 2 (Body) - outlet End - From the overall Model = 4.48381 X 10^5 lbs/radian.



$$P_{Max} = \frac{2(4.48381 \times 10^5) \cdot \cos(7^\circ)}{\left[(18.415)^2 - (18.144)^2\right]}$$

$$P_{Max} = 89{,}838.87563 \text{ psi}$$

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BY DBP DATE 1/30/78 SUBJECT Gas Storage Vessel SHEET NO 1 OF 7
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Determine Material Constant, δ

The stress distribution across a section containing a cir-

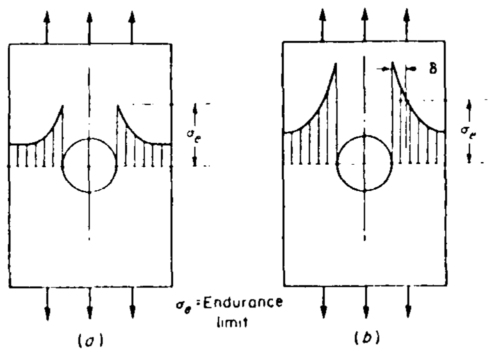


Fig. 6.29. Stress Distribution about a Circular Hole in a Bar

cular hole, Fig. 6.29*a*, has a high stress gradient at the edge of the hole. If the load is just sufficient to bring the peak stress up to the endurance limit, a fatigue failure would hardly be expected since the volume of material at this stress is zero. A finite volume of material must be at the endurance limit before a crack will form, and to obtain this volume of material the endurance limit stress must exist at some finite depth, δ, below the surface; therefore, the steeper the stress gradient, the higher the load required to produce fatigue failure, Fig. 6.29*b*.

The dimension, δ, is a property of the material; and, in general, hard, fine-grained materials have small values of δ, whereas soft, coarse-grained materials have larger values. The relationship between δ and steel tensile strengths, based on correlating fatigue data and the shear theory of failure is shown in Fig. 6.30.

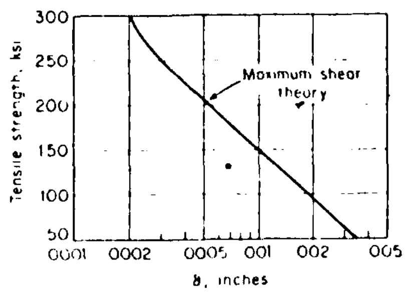


Fig. 6.30. Material Constant δ vs. Tensile Strength for Steel

For the body material, the Tensile Strength is 145 Ksi and δ is Equal to 0.00105 inches.

Calculate Stress Intensity at Depth δ

Ref: Timoshenko and Goodier, Theory of Elasticity, p. 90

The stress distribution in the vicinity of a small circular hole in the middle of a plate subjected to uniform tension is given by:

$$\sigma_r = S/2\left[1-(a/r)^2\right] + S/2\left[1+3(a/r)^4-4(a/r)^2\right]\cos 2\theta$$
$$\sigma_\theta = S/2\left[1+(a/r)^2\right] - S/2\left[1+3(a/r)^4\right]\cos 2\theta$$
$$\tau_{r\theta} = -S/2\left[1-3(a/r)^4+2(a/r)^2\right]\sin 2\theta$$

when $\theta = 0$, $\tau_{r\theta} = 0$ and the principal stresses are:

$$\sigma_r = S/2\left[2+3(a/r)^4-5(a/r)^2\right]$$
$$\sigma_\theta = S/2\left[-3(a/r)^4+(a/r)^2\right]$$

The Stress Intensity is given by:

$$S.I. = \left|\sigma_r - \sigma_\theta\right| = S/2\left[2+6(a/r)^4-6(a/r)^2\right]$$
$$= S\left[1+3(a/r)^4-3(a/r)^2\right]$$

Assume that the stress intensity distribution at the thread root radius has the same form as the above stress intensity distribution:

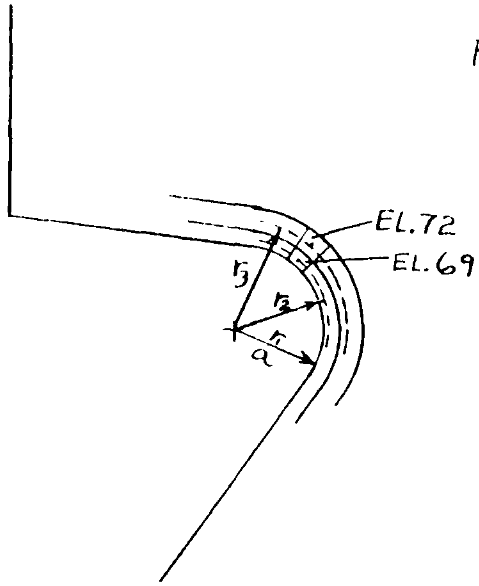
$$S.I. = S\left[1+A(a/r)^4-B(a/r)^2\right]$$

Where: a = Thread Root Radius = 0.108 in.

$r = a + \delta$, in.

δ = Distance from Surface, in.

S, A and B are three unknown constants.



From ANSYS RUN ϕDANDA8 – 11/9/78

$a = r_1 = 0.108$ in.

$r_2 = 0.118$ in.

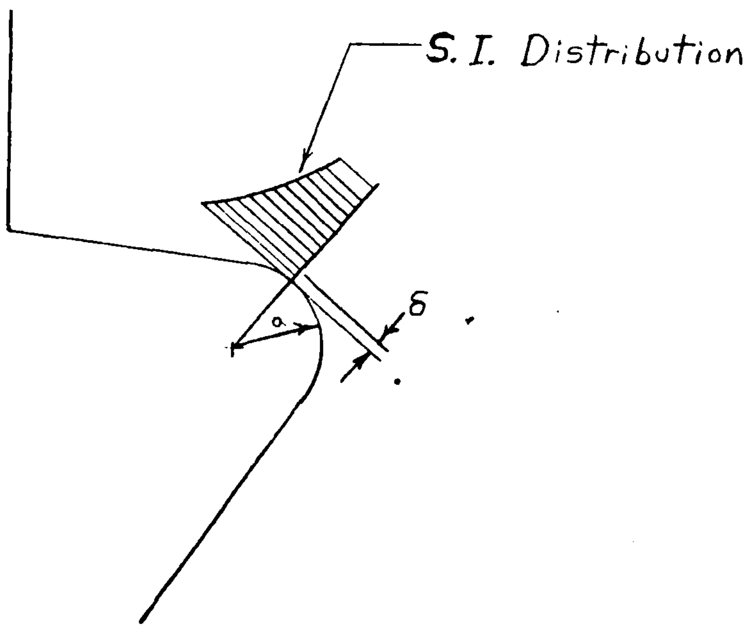
$r_3 = 0.143$ in.

At $(a/r_1) = 1$, S.I. = 222,620 psi

At $(a/r_2) = 0.91525$, S.I. = 164,832 psi

At $(a/r_3) = 0.75524$, S.I. = 94,919 psi

The Known stress Intensities at the above three Locations Can be used to evaluate the three unknowns in the stress Intensity Distribution Equation.



BY DBP DATE 11/14/78 SUBJECT Gas Storage Vessel Outlet End SHEET NO ___ OF ___
CHKD. BY ___ DATE ___ PROJ. NO JP1270

Evaluating the Constants in the Equation

$$S.I. = S\left[1 + A\left(\frac{a}{r}\right)^4 - B\left(\frac{a}{r}\right)^2\right]$$

Results in the following:

$S = 50{,}819.1065$

$A = 4.3284 \qquad B = 0.9469$

At $r = a + \delta = 0.108 + 0.00105 = 0.10905$ in.

$S.I. = 215{,}192$ psi

Therefore, the Stress Intensity at the root of Thread No. 2 on the <u>Outlet End</u> of the Body where the Thread Load is a Maximum and Equal to 4.48381×10^5 Lbs/Radian is:

$S.I.\ (Max) = 215{,}192$ psi

Fatigue Life of Threads on Outlet End Closure

$$S_{range}(Max) = 215,192 \text{ psi}$$

$$S_{alt} = 107,596 \text{ psi} \qquad S_y = 130,000 \text{ psi}$$

$$S'_{mean} = 107,596 \text{ psi} \qquad S_u = 145,000 \text{ psi}$$

$$S_{alt} + S'_{mean} > S_y \; ; \; \therefore \; S_{mean} = S_y - S_{alt}$$

$$S_{mean} = 130,000 - 107,596 = 22,404 \text{ psi}$$

$$S_{eq} = \frac{7 S_{alt}}{8 - \left[1 + \frac{S_{mean}}{S_u}\right]^3} = 116,569 \text{ psi}$$

The Design Life from the Fatigue Data from ASME Paper No. 76-PVP-62 For the body Material with a Factor of 2 on stress and a Factor of 20 on cycles is:

$$N = 680 \text{ Cycles } [\text{Design Life}]$$

Since the theoretical fatigue curves from this paper were obtained on small polished specimens tested in air, factors must be applied to account for size effects, surface finish, environmental effects, and scatter of data. Therefore, a factor of either 2 on stress or 20 on cycles, whichever is more conservative at each point, was applied to the mean failure curve to obtain a design curve which accounts for these effects. These factors have been confirmed by several fatigue tests and simulated service tests on models of components.

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BY DBP DATE 12/14/78 SUBJECT DRIVER VESSEL OUTLET END SHEET NO 1 OF 1

CHKD. BY DATE PROJ. NO JP1270

Friction Loading – 2nd Thread – Outlet End

$N = (448{,}381.)\cdot[\cos^2(7°)] = 441{,}721.584$ Lbs/Radian

$fN = 54{,}236.59074$ Lbs/Radian

$C = \frac{fN}{8} = 6{,}779.57384$ Lbs/Radian $\{f = \text{Tan } 7°\}$

$P_{Max} = 0.2003628067\, N = 88{,}504.57635$ psi

$S_{eq} = 238{,}768$ psi

For P = 47,500 psi:

N = 856 Cycles [Design Life] for P = 47,500 psi

Outlet End - With Friction

$$K = \frac{238,968}{60,000} = 3.9828$$

$U = 0.06$ {From NSWC Curve) (see page 5B-26)

Cycles Remaining on Outlet End - $P = 47,500$ psi

$$N_R = 856(1 - 0.06) = 805 \text{ cycles}$$

The detailed thread model described in Section 5.3.2 in the main body of this report, which includes the elliptical undercut on the first thread, was used to calculate the maximum stresses in the first thread. The detailed thread model described in Section 5.3.3 in the main body of this report, which has geometry typical of the second and subsequent threads, was used to calculate the maximum stresses in the threads other than the first thread. The resulting maximum stresses in threads 1, 2, 8, and 9 are shown in the following table.

Stresses in Driver Vessel Inlet End
Original Design - P = 60,000 psi

Thread No.	Thread Load (lbs/Radian)	Stress Range (psi)
1	378,073.	286,574.*
2	390,925.	380,900.
8	243,857.	210,241.
9	228,027.	190,656.

*Maximum Surface Stress Intensity from Model with Elliptical Undercut

These results indicate that the highest stress occurs in the second thread.

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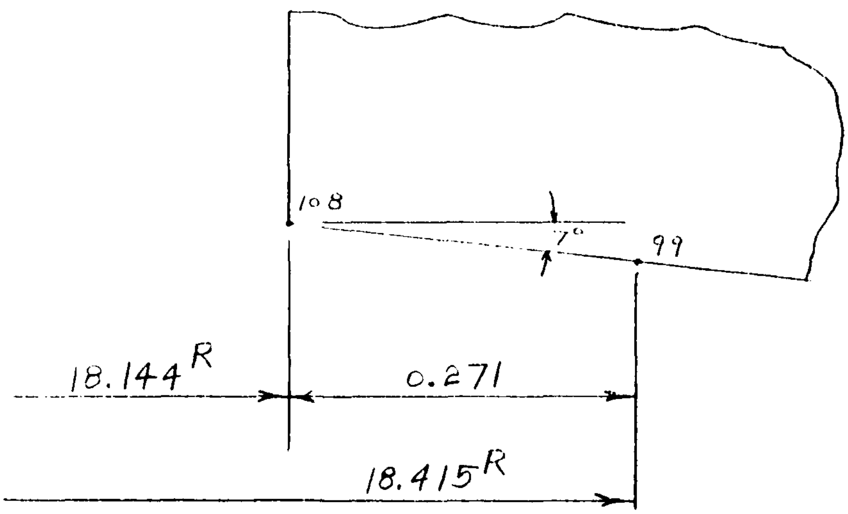
BY DBP DATE 11/14/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Maximum Equivalent Pressure on 2nd Thread

The Force on Thread No. 2 (Body) – Inlet End – From the overall Model = 390,925. Lbs/Radian.



$$P_{Max} = \frac{2(390,925.)\cdot \cos(7^\circ)}{\left[(18.415)^2 - (18.144)^2\right]}$$

$$P_{Max} = 78,326.83 \text{ psi}$$

BY DBP DATE 11/14/78 SUBJECT Gas Storage Vessel Inlet End Original Design SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP1270

From ANSYS Run ΦLJANDZI – 11/14/78 At Elements 69 and 72:

$$a = r_1 = 0.108 \text{ in}, \quad r_2 = 0.118 \text{ in}, \quad r_3 = 0.143 \text{ in}.$$

$$\text{At } \left(\frac{a}{r_1}\right) = 1, \quad S.I. = 193,741 \text{ psi}$$

$$\text{At } (a/r_2) = 0.91525, \quad S.I. = 143,434 \text{ psi}$$

$$\text{At } (a/r_3) = 0.75524, \quad S.I. = 82,534 \text{ psi}$$

Evaluating the constants in the Equation

$$S.I. = S\left[1 + A(a/r)^4 - B(a/r)^2\right]$$

Results in the following:

$$S.I. = 43,897.369\left[1 + 4.3538(a/r)^4 - 0.9403(a/r)^2\right]$$

At $r = a + \delta = 0.108 + 0.00105 = 0.10905$ in.

$$S.I. = 187,276 \text{ psi}$$

This Must be Multiplied by the following Factor to Account for the Interrupted Threads on the Inlet End:

$$S.I.(Max) = \left(\frac{60}{29.5}\right)(S.I.) \quad \left\{\begin{matrix}\text{due to } 29.5^\circ \text{ interrupted} \\ \text{Thread in every } 60^\circ \text{ Arc}\end{matrix}\right]$$

Therefore, the Stress Intensity at the root of Thread No. 2 on the Inlet End of the Body Where the thread Load is a Maximum is:

$$S.I.(Max) = \left(\frac{60}{29.5}\right)(187,276) = 380,900 \text{ psi}$$

Fatigue Life of Threads on Inlet End Closure

$$S_{range}(Max) = 380,900 \text{ psi}$$

$$S_{alt} = 190,450 \text{ psi} \qquad S_y = 130,000 \text{ psi}$$

$$S_{alt} > S_y, \therefore S_{eq} = S_{alt} = 190,450 \text{ psi}$$

The Design Life from the Fatigue Data from ASME Paper No. 76-PVP-62 For the Body Material with a Factor of 2 on Stress And a Factor of 20 on Cycles is:

$$N = 133 \text{ Cycles } [\text{Design Life}]$$

BY DBP DATE 11/21/78 SUBJECT Gas storage Vessel Inlet End original Design SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP1270

Stress in Detail Model with Friction ($f = 0.122785$)

From ANSYS Run ΦDAND4F-11/20/78 At Elements 69 and 72:

$a = r_1 = 0.108$ in., $r_2 = 0.118$ in., $r_3 = 0.143$ in.

At $\left(\frac{a}{r_1}\right) = 1$, S.I. = 215,142 psi

At $(a/r_2) = 0.91525$, S.I. = 159,423 psi

At $(a/r_3) = 0.75524$, S.I. = 90,989 psi

Evaluating the Constants in the Equation

$$S.I. = S\left[1 + A(a/r)^4 - B(a/r)^2\right]$$

Results in the following:

$$S.I. = 42{,}011.6418\left[1 + 4.8350(a/r)^4 - 0.7140(a/r)^2\right]$$

At $r = a + \delta = 0.108 + 0.00105 = 0.10905$ in.

S.I. = 208,005 psi

This Must be Multiplied by the following Factor to Account for the Interrupted Threads on the Inlet End:

$$S.I.(Max) = \left(\frac{60}{29.5}\right)(S.I.) \quad \left\{\begin{array}{l}\text{due to } 29.5^\circ \text{ interrupted} \\ \text{Thread in every } 60^\circ \text{ Arc}\end{array}\right\}$$

Therefore, the stress Intensity at the root of Thread No. 2 on the Inlet End of the Body where the Thread Load is a Maximum is:

$$S.I.(Max) = \left(\frac{60}{29.5}\right)(208{,}005) = 423{,}060 \text{ psi}$$

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BY DBP DATE 11/21/78 SUBJECT Gas storage Vessel Inlet End original Design SHEET NO 2 OF 2
CHKD. BY DATE PROJ. NO JP1270

Fatigue Life of Threads on Inlet End Closure (with Friction)

S_{range} (Max) = 423,060 psi

S_{alt} = 211,530 psi $\qquad$ S_y = 130,000 psi

$S_{alt} > S_y$, $\therefore$ $S_{eq} = S_{alt}$ = 211,530 psi

The Design Life from the Fatigue Data From ASME Paper No. 76-PVP-62 For the Body Material with a Factor of 2 on Stress And a Factor of 20 on Cycles is:

N = 100 cycles [Design Life]

Fatigue Life of Inlet End with Friction with $p = 47,500$ psi

$$S_{range}(Max) = \frac{47,500}{60,000}(423,060) = 334,923 \text{ psi}$$

$S_{alt} = 167,461$ psi $\quad S_y = 130,000$ psi

$S_{alt} > S_y$, $\therefore S_{eq} = S_{alt} = 167,461$ psi

Cycles to Failure, $N = 195$ Cycles [Design life]

If Gas Storage Vessel is operated At 47,500 psi the Fatigue Life of the Vessel will be Changed As Follows.

Inlet End

$$S_{range}(Max) = \frac{47{,}500}{60{,}000}(380{,}900) = 301{,}546 \text{ psi}$$

$S_{alt} = 150{,}773$ psi $\qquad S_y = 130{,}000$ psi

$S_{alt} > S_y$, $\therefore$ $S_{eq} = S_{alt} = 150{,}773$ psi

Cycles to Failure, $N = 270$ cycles [Design Life]

Outlet End

$$S_{range}(Max) = \frac{47{,}500}{60{,}000}(215{,}192) = 170{,}360 \text{ psi}$$

$S_{alt} = 85{,}180$ psi $\qquad S_y = 130{,}000$ psi

$S'_{mean} = 85{,}180$ psi $\qquad S_u = 145{,}000$ psi

$S_{alt} + S'_{mean} > S_y$, $\therefore$ $S_{mean} = S_y - S_{alt}$

$S_{mean} = 130{,}000 - 85{,}180 = 44{,}820$ psi

$$S_{eq} = \frac{7 S_{alt}}{8 - \left[1 + \frac{S_{mean}}{S_u}\right]^3} = 103{,}580 \text{ psi}$$

Cycles to Failure, $N = 1{,}000$ cycles [Design Life]

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BY DBP DATE 1/31/78 SUBJECT Gas Storage Vessel SHEET NO. 1 OF 2

CHKD. BY DATE PROJ. NO JPI-70

Body Material for the Gas Storage Vessel has the following Properties.

$S_U = 145{,}000$ psi

$S_y = 130{,}000$ psi

This is between Class 2 and Class 3 ASTM A-723 Material. Therefore, the Average of the Fatigue Data For Class 2 and Class 3 will be Used. Data is from ASME Paper 76-PVP-62.

Theoretical Fatigue Data

N Cycles	S_a for Class 2 psi	S_a for Class 3 psi	Average S_a psi
10	2,934,000	2,713,000	2,823,500
100	828,000	786,000	807,000
1,000	277,000	277,000	277,000
5,000	151,000	158,000	154,500
10,000	122,000	130,000	126,000
50,000	81,000	89,000	85,000
100,000	70,000	78,000	74,000
500,000	53,000	60,000	56,500
1,000,000	48,000	54,000	51,000

Fatigue Data for Body Material of Gas Storage Vessel

Factor of 2 on Stress

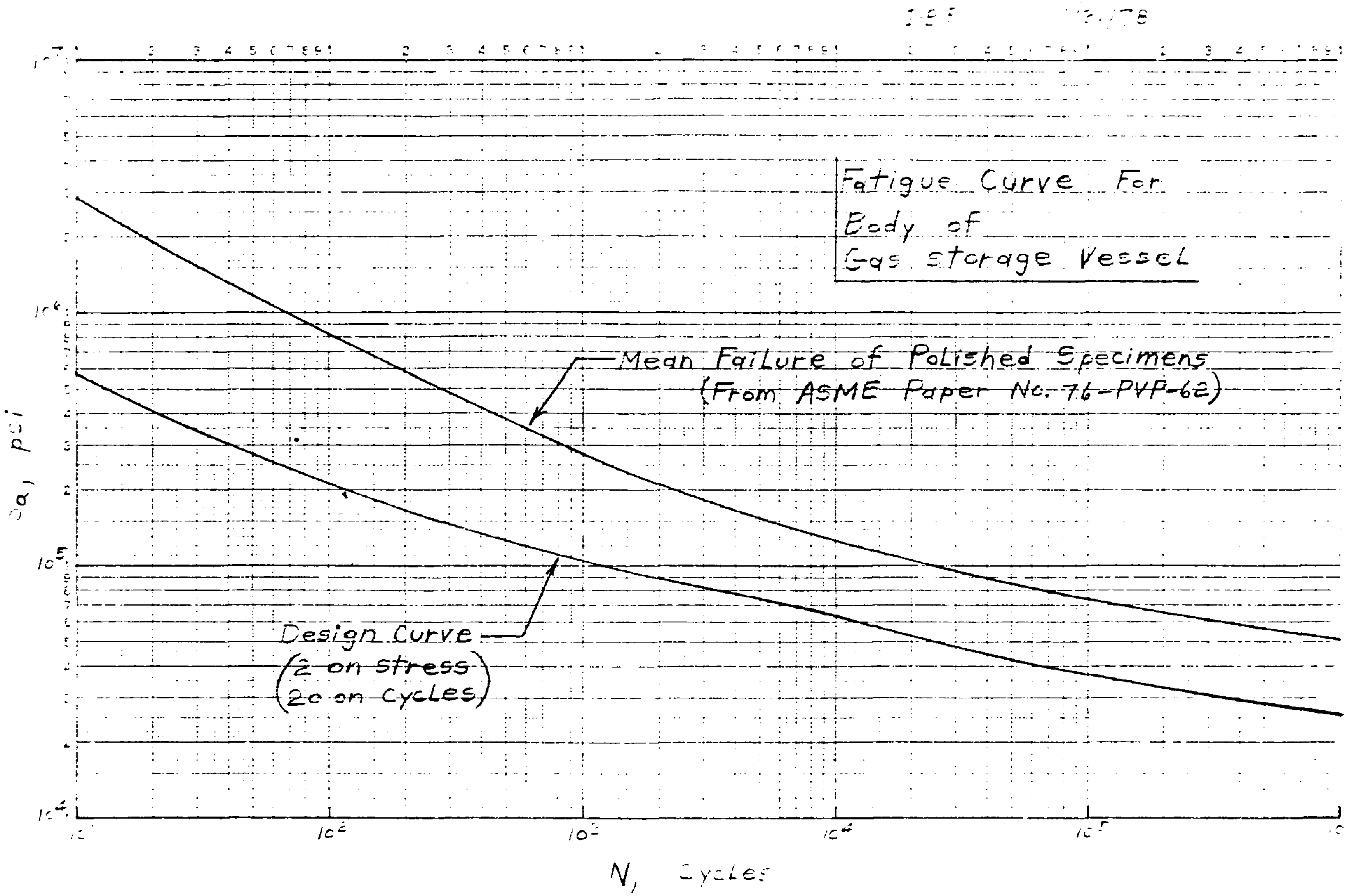
N cycles	S_a psi
10	1,411,750
100	403,500
1,000	138,500
5,000	77,250
10,000	63,000
50,000	42,500
100,000	37,000
500,000	28,250
1,000,000	25,500

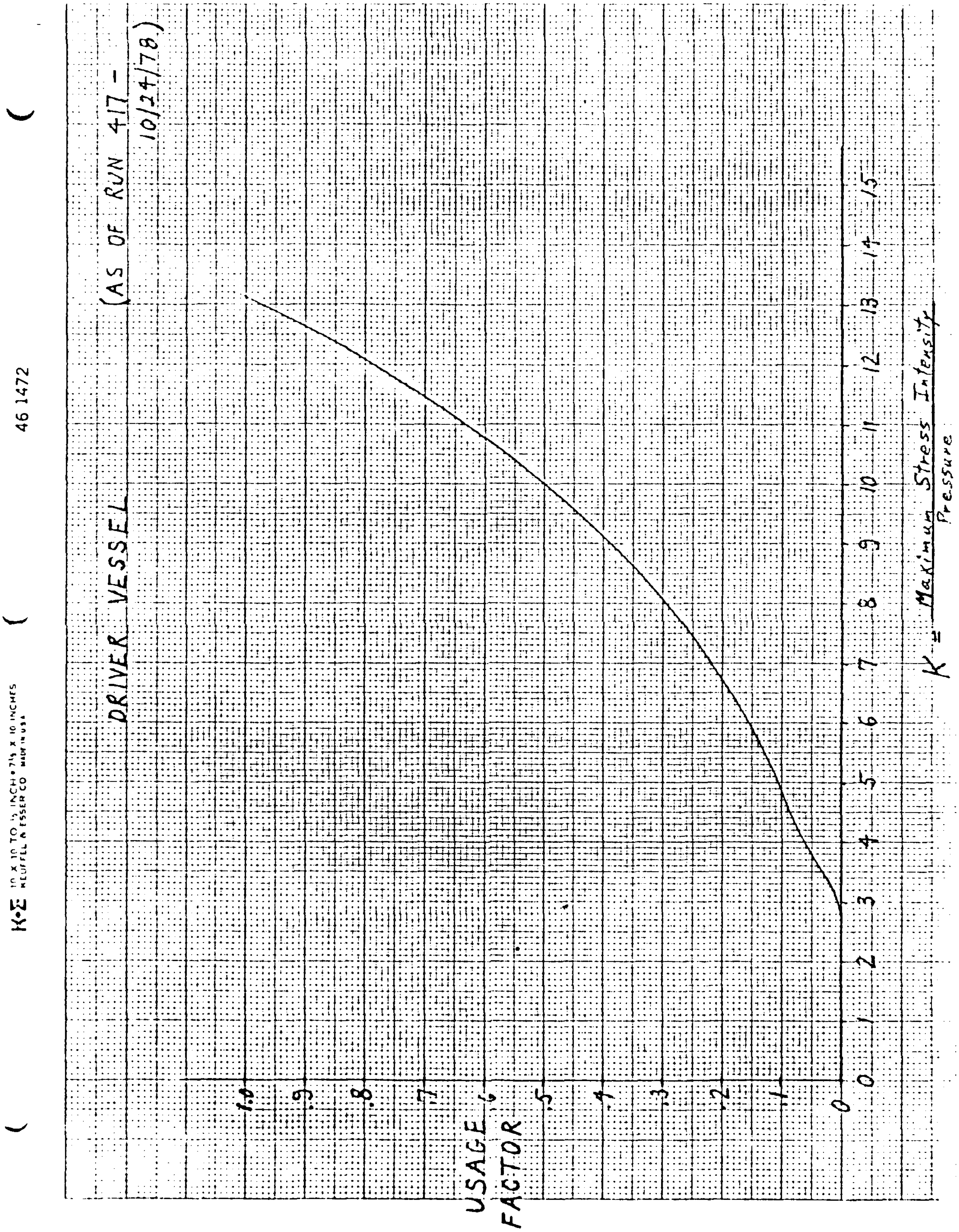
Factor of 20 on Cycles

N Cycles	S_a psi
50	277,000
250	154,500
500	126,000
2,500	85,000
5,000	74,000
25,000	56,500
50,000	51,000

Design Fatigue Curve

N Cycles	S_a psi
50	277,000
250	154,500
500	126,000
2,500	85,000
5,000	74,000
10,000	63,000
50,000	42,500
100,000	37,000
500,000	28,250
1,000,000	25,500


Fatigue Curve For
Body of
Gas storage Vessel
Mean Failure of Polished Specimens
(From ASME Paper No. 76-PVP-62)
Design Curve
(2 on stress
20 on cycles)
S_a, psi
N, Cycles
10^4
10^5
10^6
10^7
10
10^2
10^3
10^4
10^5
10^6

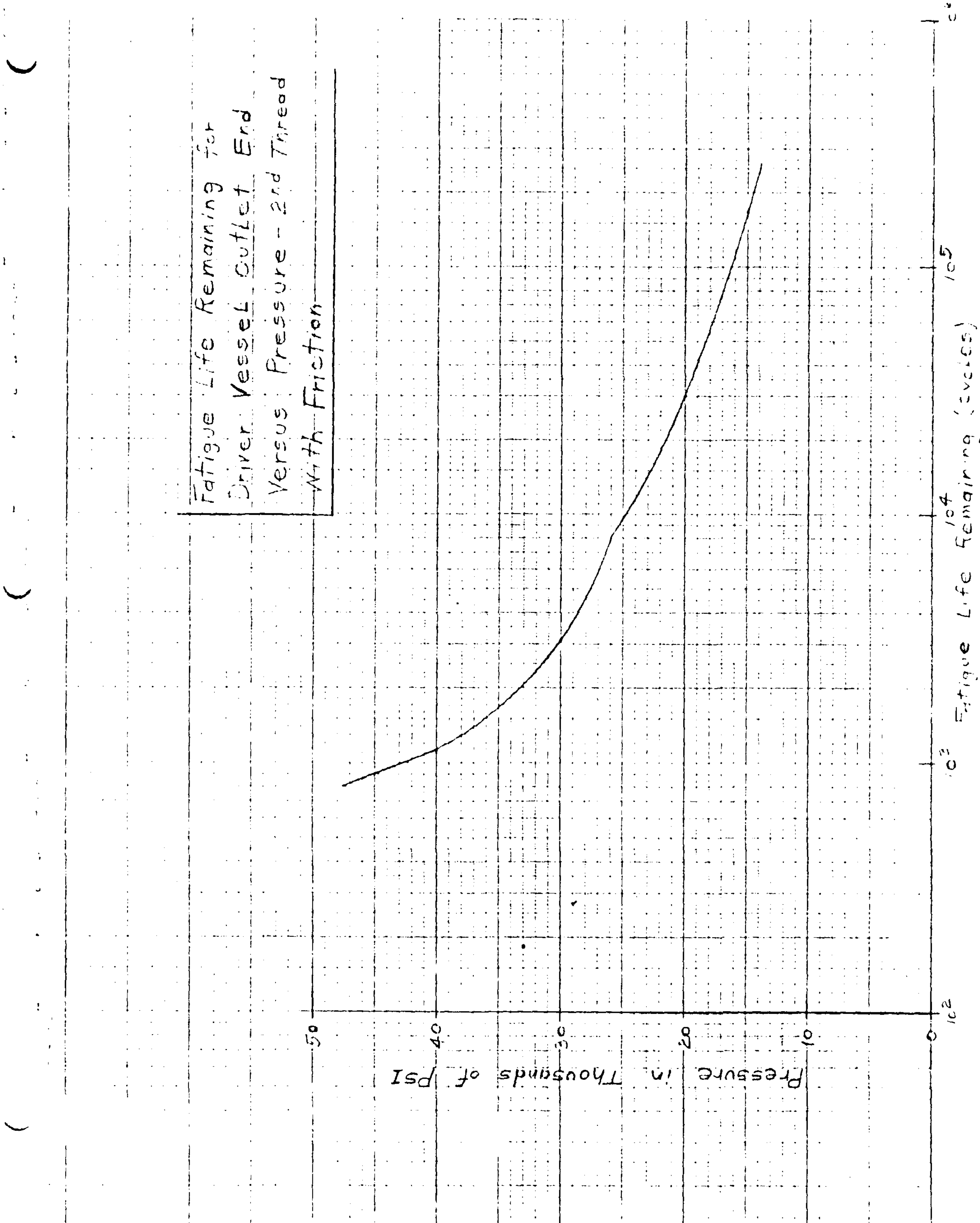

DRIVER VESSEL
(AS OF RUN 417 - 10/24/78)
USAGE FACTOR
0 .1 .2 .3 .4 .5 .6 .7 .8 .9 1.0
K = Maximum Stress Intensity / Pressure
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
K+E 10 X 10 TO ½ INCH • 7½ X 10 INCHES KEUFFEL & ESSER CO. MADE IN U.S.A.
46 1472

Fatigue Life of Driver Vessel Outlet End Vs P 2nd Thread - with Friction

P (psi)	Fatigue Life (cycles)	Fatigue Life Remaining (cycles)
47,500	856	805
45,000	952	895
40,000	1,196	1,124
30,000	3,257	3,062
25,000	10,215	9,602
20,000	31,697	29,795
15,000	176,987	166,368
26,000	8,479	7,970
24,000	12,427	11,681
22,000	19,207	18,055

N_R = Fatigue Life Remaining

$N_R = 0.94$ (Fatigue Life)



Fatigue Life Remaining for
Driver Vessel Outlet End
Versus Pressure - 2nd Thread
With Friction
Fatigue Life Remaining (cycles)
10^2
10^3
10^4
10^5
10^6
Pressure in Thousands of Psi
0
10
20
30
40
50

BY DBP DATE 12/15/78 SUBJECT DRIVER VESSEL INLET END SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP1270

Original Design - Inlet End - With Friction

(a) Thread No. 2

$$K = \frac{423,060}{60,000} = 7.051$$

$U_2^o = 0.222$ {From NSWC Curve)

(b) Thread No. 8

$$K = \frac{237,335}{60,000} = 3.9556$$

$U_8^o = 0.06$ {From NSWC Curve)

Cycles Remaining For Original Design - With Friction

$$N_R = 195(1-0.222) = 152 \text{ cycles}$$

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BY DBP DATE 12/15/78 SUBJECT Driver Vessel Original Design

SHEET NO 1 OF 1

CHKD. BY DATE

PROJ. NO JP1270

Fatigue Life of Threads on Driver Vessel For Pressure of 60,000 psi

LOCATION	Stress Range, psi	Fatigue Design Life
Outlet End (No Friction)	215,192	680 cycles
Outlet End with Friction	238,968	532 cycles
Inlet End (No Friction)	380,900	133 cycles
Inlet End with Friction	423,060	100 cycles

Fatigue Life of Threads on Driver Vessel For Pressure of 47,500 psi

LOCATION	Stress Range, psi	Fatigue Design Life
Outlet End (No Friction)	170,360	1,000 cycles
Outlet End with Friction	189,183	856 cycles
Inlet End (No Friction)	301,546	270 cycles
Inlet End with Friction	334,923	195 cycles

APPENDIX 5C

FRACTURE MECHANICS EVALUATION OF THREADS

for

DRIVER VESSEL

ORIGINAL DESIGN

Crack Growth Rate Analysis of Threads on the Gas Storage Vessel:

REFERENCES:

(1) Imhof, E. J. and Barsom, J. M., "Fatigue and Corrosion-Fatigue Crack Growth of 4340 Steel At Various Yield Strengths", Progress in Flaw Growth and Fracture Toughness Testing, ASTM STP 536, American Society for Testing and Materials, 1973, pp. 182-205.

(2) Wessel, E. T. and Mager, T. R., "Fracture Mechanics Technology As Applied to Thick-Walled Nuclear Pressure Vessels", Proc. Conf. on Practical Application of Fracture Mechanics to Pressure Vessel Technology, Institution of Mechanical Engineers, 1971.

BASIC ASSUMPTIONS

1. Thread Material is modified AISI 4340, or "gun steel." This is now designated ASTM A-723 Material. Assume this Material has the following Properties:

$$S_U = 145{,}000 \text{ psi}$$

$$S_Y = 130{,}000 \text{ psi}$$

$$K_{Ic} = 100 \text{ Ksi}\sqrt{in}$$

2. From Reference (1), the crack growth rate for this material is represented by the following Equation:

$$\frac{da}{dN} = 0.66 \times 10^{-8} (\Delta K)^{2.25}$$

Where: $\frac{da}{dN}$ = Crack Growth Rate, inches/cycle

ΔK = Stress Intensity Factor Range, $\text{Ksi}\sqrt{in}$

3. Assume there is a thin Surface defect oriented normal to the Maximum Surface Stress At the inside Surface of the thread root radius where the Maximum stress occurs.

4. Assume that the stress Range is Equal to the Maximum Surface Stress.

Procedure given in Reference (2) will be followed:

1. The Fracture Toughness, K_{IC}, is:

$$K_{IC} = 100 \text{ Ksi}\sqrt{\text{in}}$$

2. From Reference (1), the Crack Growth Rate, da/dN, is:

$$\frac{da}{dN} = C_o \Delta K^n$$

$$\frac{da}{dN} = 0.66 \times 10^{-8} (\Delta K)^{2.25} \quad \left\{ \text{For 4340 Mat'L from Ref. (1)} \right]$$

where: $\frac{da}{dN}$ = Crack Growth Rate, inches/cycle

C_o = Empirical intercept Constant

ΔK = Stress Intensity Factor Range, Ksi$\sqrt{\text{in}}$

n = Slope of da/dN Versus Log ΔK Curve

Procedure (continued)

The Crack Growth Rate Equation From Reference (1) is shown in the curve below. Note that the Equation is an upper bound of the plotted data.

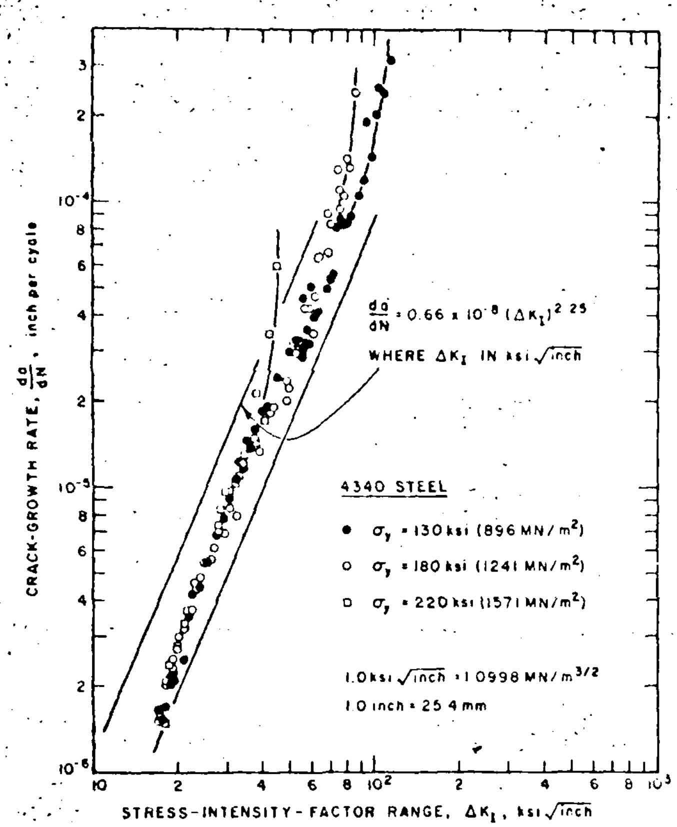


FIG. 9–Fatigue-crack growth in 4340 steel of various yield strengths

Procedure (continued)

3. For a thick-walled Pressure Vessel Containing a thin ($a/\ell \approx 0$) Surface defect oriented normal to the Maximum Surface Stress, the Critical Crack depth, a_{cr}, is:

$$a_{cr} \cong \frac{K_c^2}{1.25\pi\sigma^2} \qquad \left\{\text{Minimum } a_{cr}\right)$$

Where: a_{cr} = Critical Crack Depth, inches

K_c = Fracture Toughness, $Ksi\sqrt{in}$

σ = Maximum Surface Stress, Ksi

4. The Number of Cycles to grow to Critical Flaw size (failure), N, is:

$$N = \frac{2}{(n-2)C_o M^{n/2} \Delta\sigma^n} \left(\frac{1}{a_i^{(n-2)/2}} - \frac{1}{a_{cr}^{(n-2)/2}} \right)$$

Where: N = Number of Cycles to Failure

a_i = Initial Crack Depth, inches

n = Slope of da/dN Versus Log ΔK Curve

a_{cr} = Critical Crack Depth, inches

C_o = Empirical intercept Constant for ΔK in $psi\sqrt{in}$

$\Delta\sigma$ = Applied cyclic stress Range, psi

$M = 1.25\pi$

Threads on Outlet End Closure

If $\sigma = \Delta\sigma = 215{,}192$ psi

1. $K_{Ic} = 100$ Ksi $\sqrt{in}$

2. Critical Crack Depth

$$a_{cr} = \frac{1}{1.25\,\pi}\left(\frac{100{,}000}{215{,}192}\right)^2 = 0.054990''$$

3. Cycles to Failure

$C_o = 1.17366 \times 10^{-15}$ for ΔK in psi $\sqrt{in}$

$(n-2) = 0.25 \qquad M^{n/2} = (1.25\pi)^{1.125} = 4.659264564$

$\Delta\sigma^n = (215{,}192)^{2.25} = 9.973756802 \times 10^{11}$

$$\frac{1}{a_{cr}^{(n-2)/2}} = \frac{1}{(0.054990)^{0.125}} = 1.4370$$

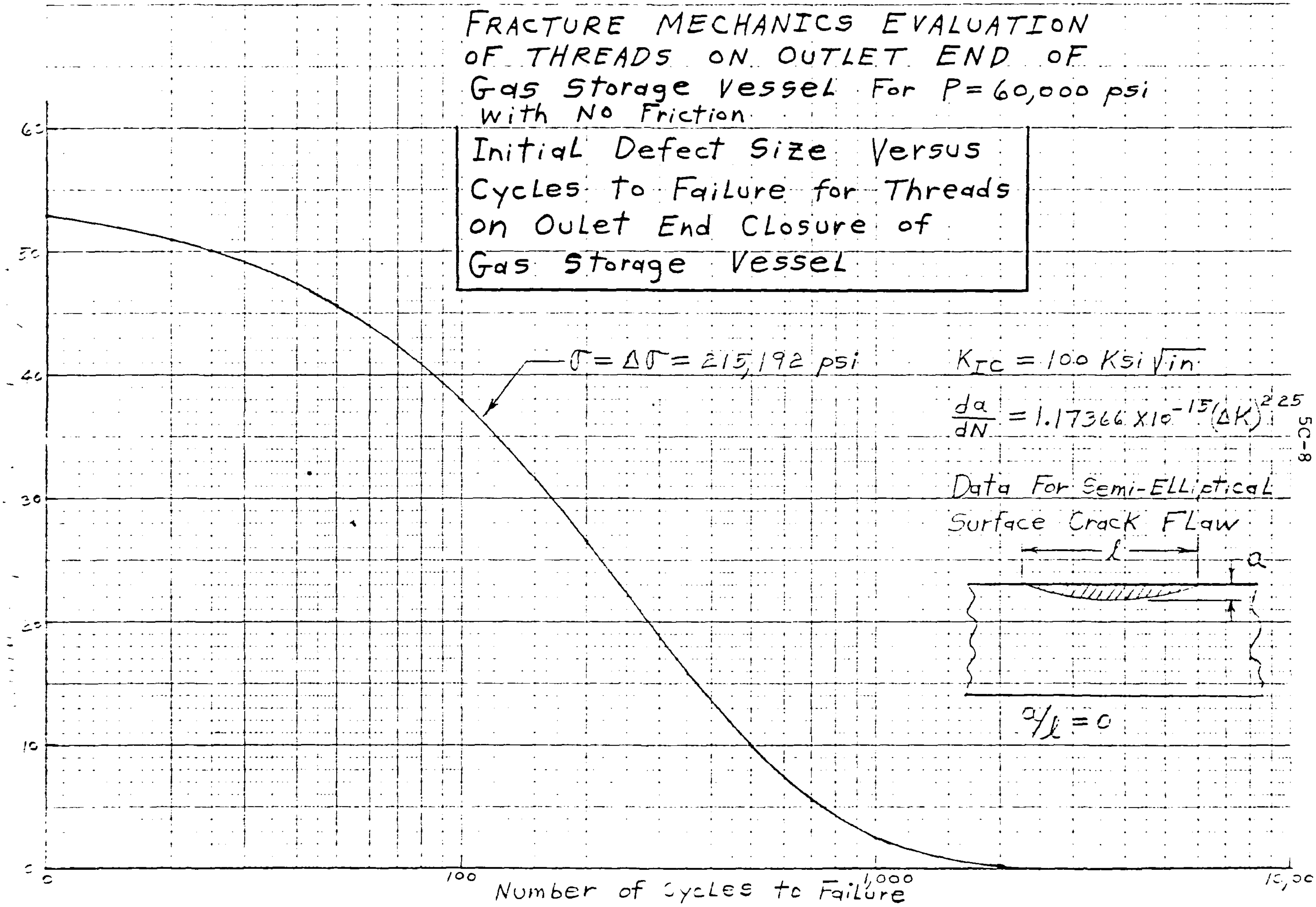
$$N = 1466.798759\left[\frac{1}{a_i^{0.125}} - 1.43702436\right]$$

$$a_i = \left(\frac{1466.798759}{N + 2107.825536}\right)^8.$$

a_i Versus N for Threads on Outlet End Closure
$\sigma = \Delta\sigma = 215{,}192$ psi, $K_{IC} = 100$ Ksi$\sqrt{in}$
Modified AISI 4340 Material

a_i inches	N Cycles
0.052947	10
0.050989	20
0.045586	50
0.037953	100
0.026628	200
0.010017	500
0.005546	700
0.002462	1000
0.0002643	2000
0.000003289	5000

$$a_i = \left(\frac{1466.798759}{N + 2107.825536}\right)^8$$



FRACTURE MECHANICS EVALUATION OF THREADS ON OUTLET END OF Gas Storage Vessel For P=60,000 psi with No Friction
Initial Defect Size Versus Cycles to Failure for Threads on Oulet End Closure of Gas Storage Vessel
σ = Δσ = 215,192 psi
K_IC = 100 Ksi √in
da/dN = 1.17366 X10^-15 (ΔK)^2.25
Data For Semi-ELLiptical Surface Crack FLaw
ℓ
a
a/ℓ = 0
60
50
40
30
20
10
0
0
100
1,000
10,000
Number of Cycles to Failure

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BY DBP DATE 12/19/78 SUBJECT Driver Vessel Outlet End SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP1270

Outlet End - 2nd Thread - with Friction - P = 45,000 psi

If $\sigma = \Delta\sigma = 179,226$ psi and $K_{IC} = 100$ ksi$\sqrt{in}$

1. $K_{IC} = 100$ ksi$\sqrt{in}$

2. Critical Crack Depth

$$a_{cr} = \frac{1}{1.25\pi}\left(\frac{100,000}{179,226}\right)^2 = 0.079275''$$

3. Cycles to Failure

$C_o = 1.17366 \times 10^{-15}$ for ΔK in psi$\sqrt{in}$

$(n-2) = 0.25 \qquad M^{n/2} = (1.25\pi)^{1.125} = 4.659264564$

$\Delta\sigma^n = (179,226)^{2.25} = 6.609251488 \times 10^{11}$

$$\frac{1}{a_{cr}^{(n-2)/2}} = \frac{1}{(0.079275)^{0.125}} = 1.37280148$$

$$N = 2,213.478814\left[\frac{1}{a^{0.125}} - 1.3728\right]$$

$$a_i = \left(\frac{2,213.487814}{N + 3,038.680046}\right)^8$$

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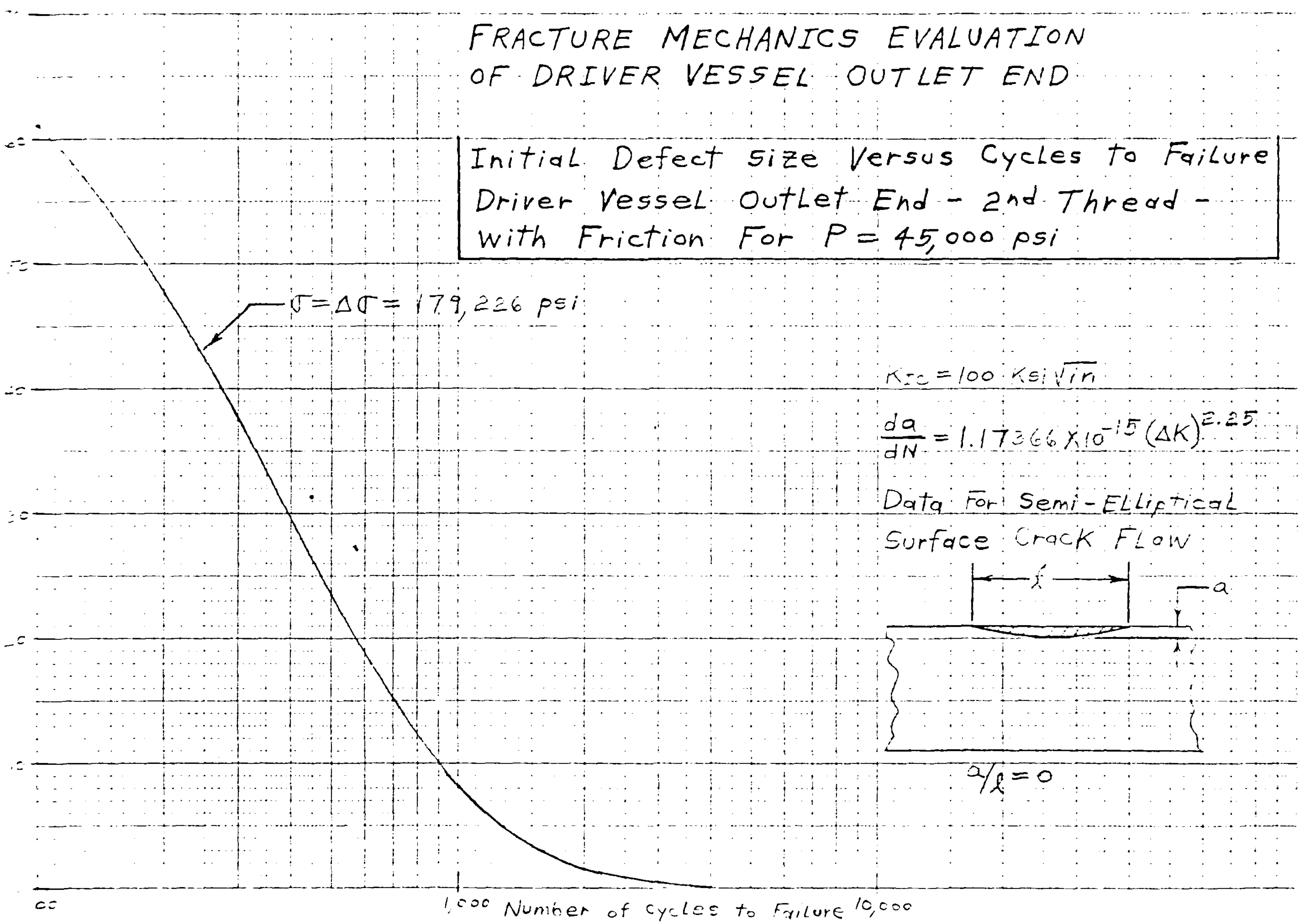
BY DBP DATE 12/19/78 SUBJECT Driver Vessel Outlet End SHEET NO 2 OF 2
CHKD. BY DATE PROJ. NO JP1270

Driver Vessel Outlet End – 2nd Thread – With Friction for P = 45,000 psi

a_i Versus N for Threads on Outlet End Closure
$\sigma = \Delta\sigma = 179{,}226$ psi, $K_{IC} = 100$ Ksi$\sqrt{in}$
Modified AISI 4340 Material

a_i inches	N Cycles
0.07721867	10
0.07522177	20
0.06957187	50
0.06118440	100
0.04760721	200
0.03732662	300
0.02343617	500
0.01509631	700
0.00814144	1,000
0.00138702	2,000
0.000325897	3,000
0.00009565	4,000
0.0000330475	5,000
0.0000129353	6,000

$$a_i = \left(\frac{2{,}213.487814}{N + 3{,}038.680046}\right)^8$$


FRACTURE MECHANICS EVALUATION
OF DRIVER VESSEL OUTLET END
Initial Defect size Versus Cycles to Failure
Driver Vessel Outlet End - 2nd Thread -
with Friction For P = 45,000 psi
$\sigma = \Delta\sigma = 179,226$ psi
$K_{IC} = 100$ Ksi$\sqrt{in}$
$\frac{da}{dN} = 1.17366 \times 10^{-15} (\Delta K)^{2.25}$
Data For Semi-Elliptical
Surface Crack Flaw
ℓ
a
$a/\ell = 0$
1,000
Number of cycles to Failure
10,000

Threads on Inlet End Closure

IF $\sigma = \Delta\sigma = 380{,}900$ psi and $K_{IC} = 100$ Ksi$\sqrt{in}$

1. $K_{IC} = 100$ Ksi$\sqrt{in}$

2. Critical Crack Depth

$$a_{cr} = \frac{1}{1.25\pi}\left(\frac{100{,}000}{380{,}900}\right)^2 = 0.0175517''$$

3. Cycles to Failure

$C_o = 1.17366 \times 10^{-15}$ for ΔK in psi$\sqrt{in}$

$(n-2) = 0.25$ $\qquad M^{n/2} = (1.25\pi)^{1.125} = 4.659264564$

$\Delta\sigma^n = (380{,}900)^{2.25} = 3.604331177 \times 10^{12}$

$$\frac{1}{a_{cr}^{(n-2)/2}} = \frac{1}{(0.01755169)^{0.125}} = 1.65753$$

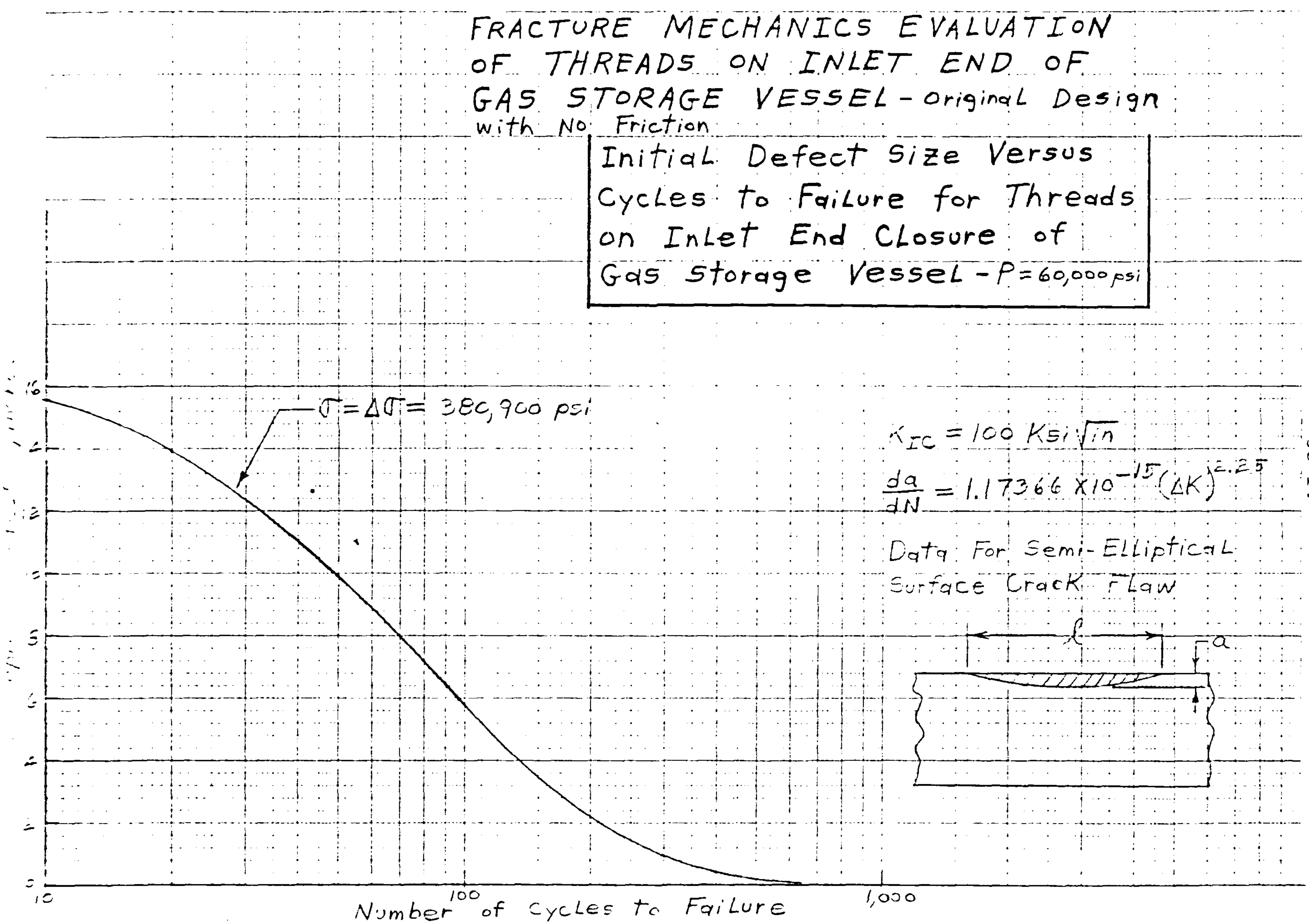
$$N = 405.888147\left[\frac{1}{a_i^{0.125}} - 1.65753\right]$$

$$a_i = \left(\frac{405.888147}{N + 672.769842}\right)^8.$$

a_i Versus N for Threads on
Inlet End Closure
$\sigma = \Delta\sigma = 380{,}900$ psi, $K_{IC} = 100$ Ksi $\sqrt{in}$
Modified AISI 4340 Material

a_i inches	N Cycles
0.015548	10
0.013885	20
0.009891	50
0.005792	100
0.002188	200
0.0009187	300
0.00041995	400
0.00020585	500
0.00010697	600

$$a_i = \left(\frac{405.888147}{N + 672.769842}\right)^8.$$


FRACTURE MECHANICS EVALUATION
OF THREADS ON INLET END OF
GAS STORAGE VESSEL - original Design
with No Friction
Initial Defect Size Versus
Cycles to Failure for Threads
on Inlet End Closure of
Gas Storage Vessel - P=60,000 psi
σ=Δσ= 380,900 psi
K_IC = 100 Ksi √in
da/dN = 1.17366 X10^-15 (ΔK)^2.25
Data For Semi-Elliptical
Surface Crack Flaw
ℓ
a
10
100
1,000
Number of Cycles to Failure

APPENDIX 6A

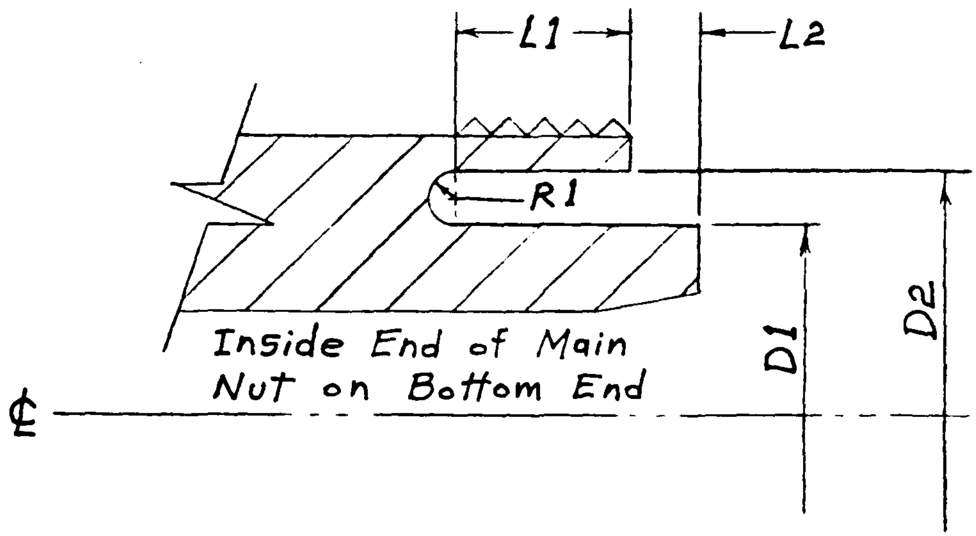
DESIGN MODIFICATIONS TO
MACH 14/18 HEATER VESSEL

BY DBP DATE 1/15/79 SUBJECT M14/18 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Summary of Fatigue Life Remaining on Modified M14/18 Heater Vessel Bottom End Based on P= 28,000 psi
(Elliptical Undercut Not Taken Into Account)

DESIGN	L1 (inches)	L2 (inches)	D1 (inches)	D2 (inches)	Critical Thread No.	Life Remaining No Friction
Original	0	0	—	—	1	0
REV. 1*	3 1/2	1/2	27 1/2	29	1	0
REV. 2*	4	3	27 1/2	29	1	0

* R1 = 3/8"



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BY DBP DATE 1/15/79 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

M14/18 Heater Vessel Bottom End
Original Design - P = 46,000 psi

Thread No.	Thread Load (Lbs/Radian)	Stress Range (psi)
1	356,468.	765,532.
2	352,199.	378,338.
4	265,715.	281,467.
7	200,350.	202,242.
10	154,211.	144,921.

M 14/18 Heater Vessel Bottom End
REV. 1 Design - P = 46,000 psi

Thread No.	Thread Load (Lbs/Radian)	Stress Range (psi)
1	61,208.4	568,608.
7	270,307.	312,939.

M 14/18 Heater Vessel Bottom End
REV. 2 Design - P = 46,000 psi

Thread No.	Thread Load (Lbs/Radian)	Stress Range (psi)
1	0	487,929.
4	115,753.2	300,948.
10	270,066.	311,326.

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BY DBP DATE 2/7/79 SUBJECT M14/18 Heater Vessel Bottom End SHEET NO 1 OF 1

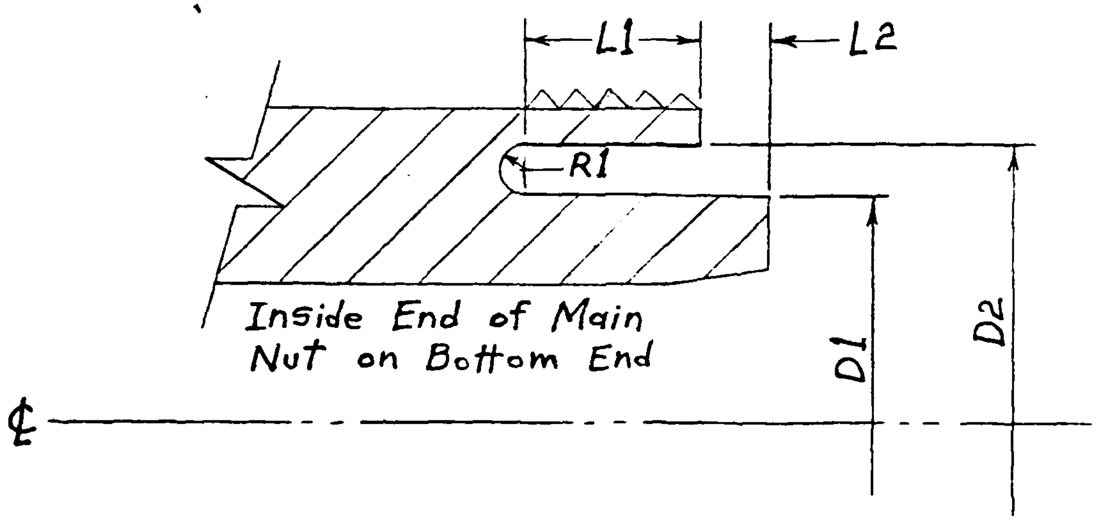
CHKD. BY DATE PROJ. NO JP/270

Summary of Fatigue Life Remaining on Modified M 14/18 Heater Vessel Bottom End Based on P = 28,000 psi

DESIGN	L1 (inches)	L2 (inches)	D1 (inches)	D2 (inches)	Critical Thread No.	Life Remaining No Friction	Life Remaining with Friction
Original	0	0	—	—	2	422 Cycles	303 Cycles
REV. 2*	4	3	27 1/2	29	4	775 Cycles	721 Cycles

*R1 = 3/8"

Note: With Friction, A Coefficient of Friction, f, of f = 0.12278 Was used.



BY DBP DATE 2/5/79 SUBJECT M14/18 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

M 14/18 Heater Vessel Bottom End
Original Design - P = 46,000 psi

Thread No.	Load (Lbs/Radian)	Stress Range (psi)
1	356,468.	308,628.*
2	352,199.	378,338.
4	265,715.	281,467.
7	200,350.	202,242.
10	154,211.	144,921.

*Maximum Surface Stress Intensity From Model with Elliptical Undercut.

M14/18 Heater Vessel Bottom End
REV. 2 Design - P = 46,000 psi

Thread No.	Load (Lbs/Radian)	Stress Range (psi)
1	0	119,547.*
2	0	168,191.
4	115,753.2	300,948.
10	270,066.	311,326.

*Maximum Surface Stress Intensity From Model with Elliptical Undercut.

Current Usage Factor For M14 Heater Vessel Bottom End

(a) Thread No. 2

$$K = \frac{378,338}{46,000} = 8.2247$$

$U_2^{\circ} = 0.265$ (From curve on page 6A-27)

Cycles Remaining For P = 28,000 psi

$$N_R = 575(1-0.265) = 422 \text{ cycles}$$

Original Design-Bottom End - with Friction

Thread No. 2

$$K = \frac{422,332}{46,000} = 9.1811$$

$U_2^{\circ} = 0.333$ (From NSWC Curve) (see page 6A-27)

Cycles Remaining For Original Design-With Friction For P = 28,000 psi

$$N_R = 455(1-0.333) = 303 \text{ cycles}$$

BY DBP DATE 2/5/79 SUBJECT M14/18 Heater Vessel Bottom End SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP1270

Current Usage Factor For M14 Heater Vessel Bottom End

(a) Thread No. 2

$$K = \frac{378,338}{46,000} = 8.2247$$

$U_2^{\circ} = 0.265$ {From NSWC Curve)

(b) Thread No. 4

$$K = \frac{281,467}{46,000} = 6.1188$$

$U_4^{\circ} = 0.15$ {From NSWC Curve)

(c) Thread No. 10

$$K = \frac{144,921}{46,000} = 3.1505$$

$U_{10}^{\circ} = 0$ {From NSWC Curve)

BY DBP DATE 2/5/79 SUBJECT M14/18 Heater Vessel Bottom End SHEET NO 2 OF 2
CHKD. BY DATE PROJ. NO JP1270

Cycles Remaining For REV. 2 Design For P = 28,000 psi

$$U_2 = 0.265 + \frac{N_R}{8,950} \quad \text{(Second Thread)}$$

$$U_4 = 0.15 + \frac{N_R}{912} \quad \text{(Fourth Thread)}$$

$$U_{10} = 0 + \frac{N_R}{853} \quad \text{(Tenth Thread)}$$

By setting U_2, U_4 and U_{10} Equal to 1.0, N_R For Each Thread is determined:

(a) For Thread No. 2:

$$N_R = 8,950(1 - 0.265) = 6,578 \text{ cycles}$$

(b) For Thread No. 4:

$$N_R = 912(1 - 0.15) = 775 \text{ cycles}$$

(c) For Thread No. 10:

$$N_R = 853 \text{ cycles}$$

The smallest value of N_R must be used.

Therefore, the cycles Remaining for the REV. 2 Design is 775 cycles.

BY DBP DATE 2/7/79 SUBJECT M14/18 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP/270

Original Design - Bottom End - With Friction

Thread No. 4

$$K = \frac{314,659}{46,000} = 6.8404$$

$$U_4^{\circ} = 0.18 \quad \{\text{From NSWC Curve}\}$$

Cycles Remaining For REV. 2 Design - With Friction
(4th Thread) - For $P = 28,000$ psi

$$N_R = 879(1 - 0.18) = 721 \text{ Cycles}$$

Friction Loading – 4th Thread – REV. 2 Design

$N = 115{,}753.2 \cdot [\cos^2(7°)] = 114{,}034.0176$ Lbs/Radian

$fN = 14{,}001.61678$ Lbs/Radian $\{f = \tan 7°\}$

$C = \frac{fN}{8} = 1{,}750.2021$ Lbs/Radian

$P_{Max} = 0.231191259\,N = 26{,}363.67$ psi

Friction Loading – 4th Thread – original Design

$N = 265{,}715. [\cos^2(7°)] = 261{,}768.5645$ Lbs/Radian

$fN = 32{,}141.13824$ Lbs/Radian

$C = \frac{fN}{8} = 4{,}017.6423$ Lbs/Radian

$P_{Max} = 0.231191259\,N = 60{,}518.604$ psi

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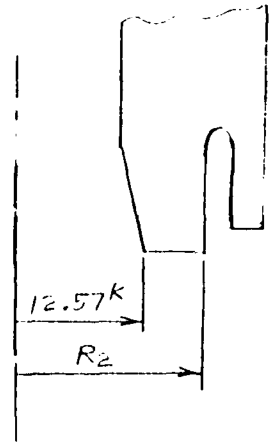
BY DBP DATE 12/22/78 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 1 OF 4

CHKD BY DATE PROJ. NO JP1270

Bearing stress on Nut

For $P = 35{,}000$ psi, The End Load is:

$$F = \frac{\pi}{4}(24)^2(35{,}000) = 15{,}833{,}626.97 \text{ Lbs}$$



The Bearing stress is:

$$S_b = \frac{F}{\pi\left[R_2^2 - (12.57)^2\right]}$$

The Bearing stress Must be Less than $S_y = 130{,}000$ psi, therefore:

$$S_y \geq \frac{F}{\pi\left[R_2^2 - 158.0049\right]}$$

$$R_2 \geq \sqrt{\frac{F}{\pi S_y} + 158.0049}$$

$$R_2 \geq \sqrt{\frac{15{,}833{,}626.97}{\pi(130{,}000)} + 158.0049}$$

$$R_2 \geq 14.02762''$$

ENGINEERING DESIGN & ANALYSIS SERVICES

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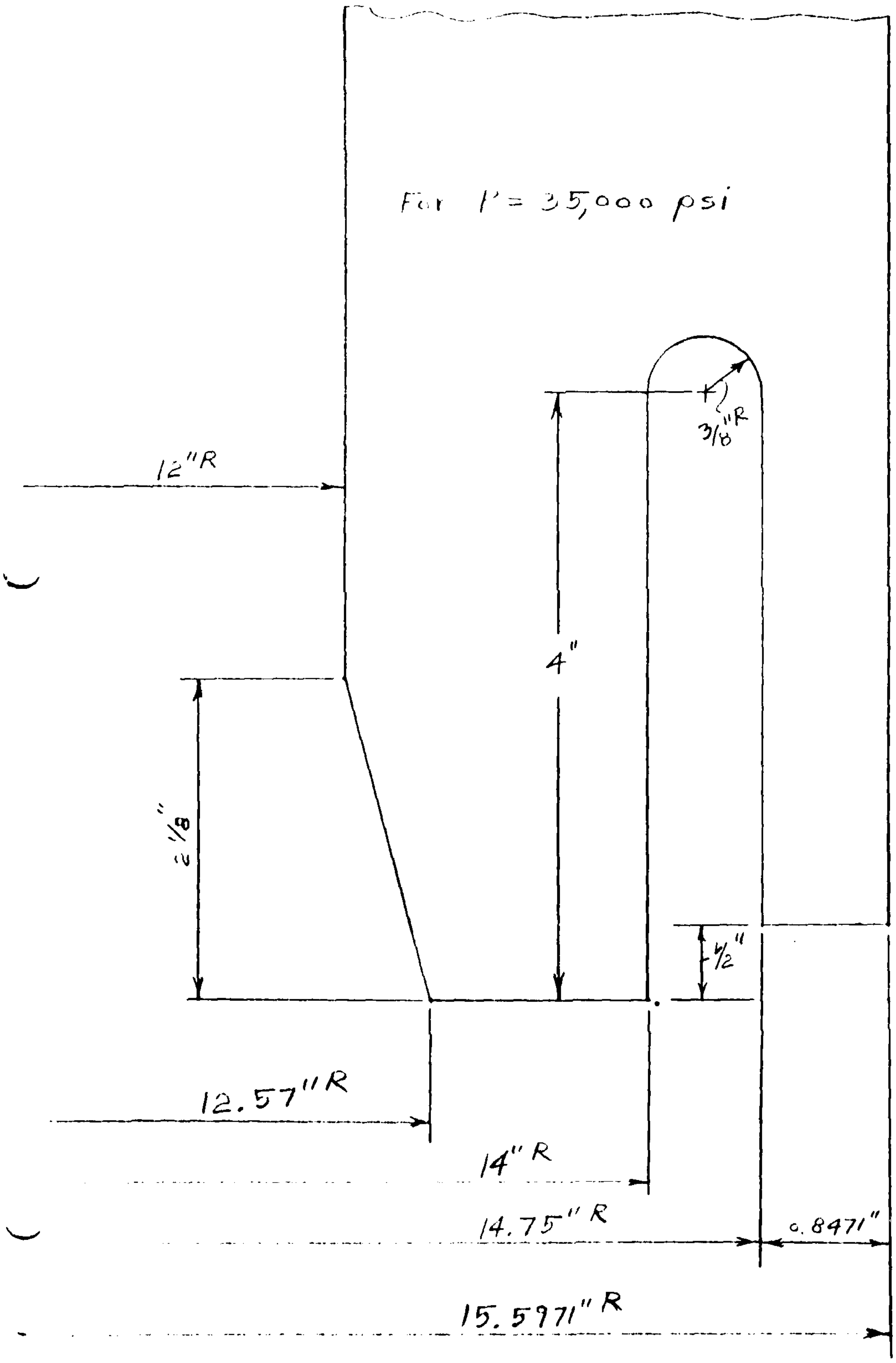
BY DBP DATE 12/22/78 SUBJECT M14/18 Heater Vessel Bottom End SHEET NO 2 OF 4
CHKD. BY DATE PROJ. NO JP1270



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HYPERVELOCITY WIND TUNNEL COMPONENTS STRUCTURAL EVALUATION. VOL--ETC(U)
MAY 79 D PETERSON , E WESTERMANN N60921-78-C-0013
UNCLASSIFIED ODAI-1270-8-79-VOL-2 NL

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For $P = 30{,}000$ psi :

$$F = \frac{\pi}{4}(24)^2(30{,}000) = 13{,}571{,}680.26 \text{ lbs}$$

$$R_2 \geq \sqrt{\frac{13{,}571{,}680.26}{\pi(130{,}000)} + 158.0049}$$

$$R_2 \geq 13.8288''$$

For $P = 28{,}000$ psi :

$$F = \frac{\pi}{4}(24)^2(28{,}000) = 12{,}666{,}901.58$$

$$R_2 \geq 13.748'' \Rightarrow \text{use } R_2 = 13.75''$$

$$\sigma_{bearing} = \frac{12{,}666{,}901.58}{\pi\left[(13.75)^2 - (12.57)^2\right]} = 129{,}823.3 \text{ psi}$$

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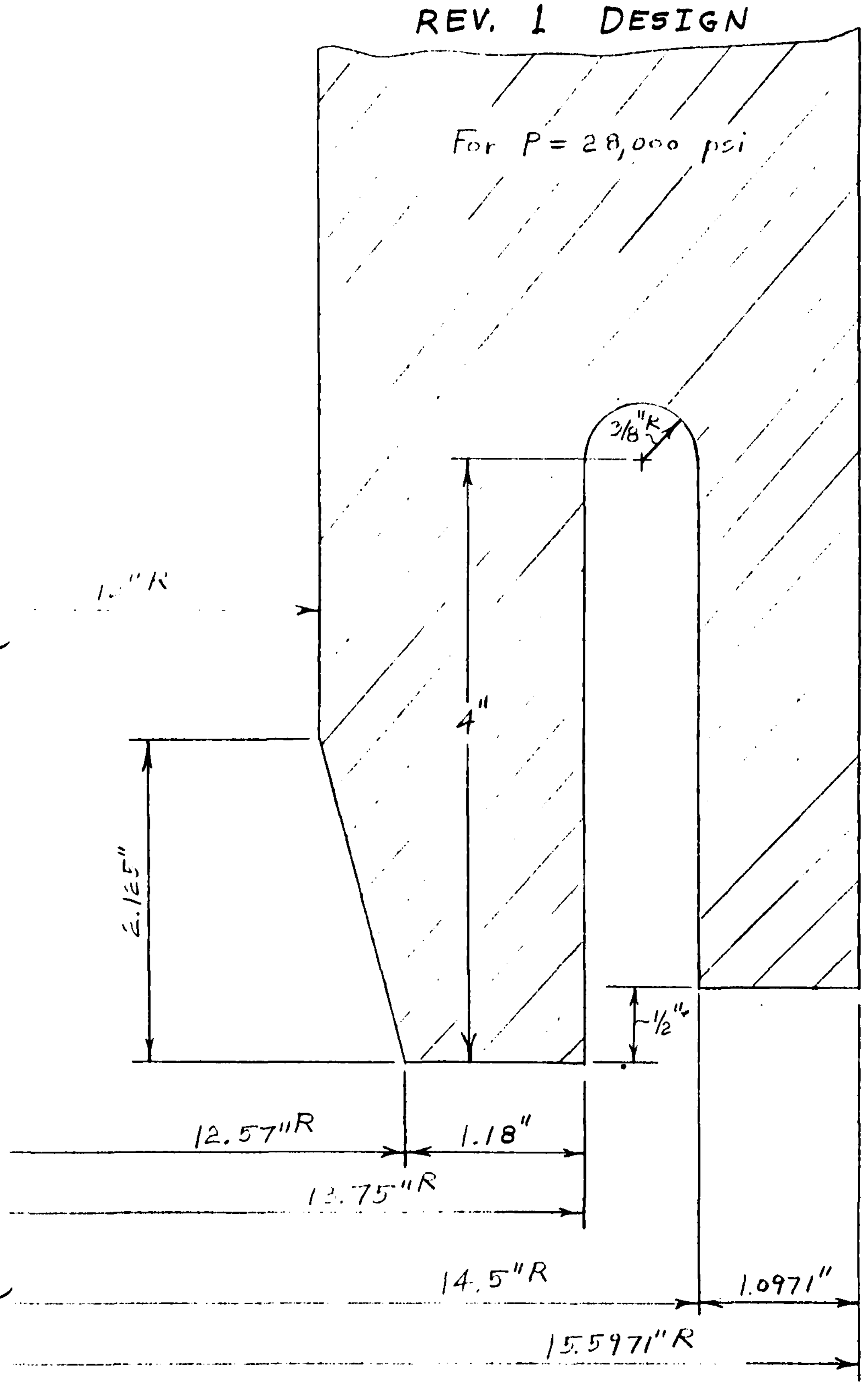
BY DBP DATE 12/22/78 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 4 OF 4
CHKD. BY DATE PROJ. NO JP/270

REV. 1 DESIGN



BY DBP DATE 12/27/78 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

REV. 1 DESIGN

Node Coordinates

Node	X (in)	Y (in)
1752	13.75	60.0
1753	14.50	60.5
1770	15.4721	60.5
1771	15.5971	60.5
1772	15.802033	60.5
1773	15.8652/743	60.5
1792	13.75	61.0
1793	14.5	61.0
1832	13.75	62.0
1833	14.5	62.0
1872	13.75	63.0
1873	14.5	63.0
1912	13.75	64.0
1913	14.5	64.0
3200	13.85983496	64.26516504
3201	14.125	64.375
3202	14.39016504	64.26516504
3203	13.75	64.3
3204	13.75	64.75
3205	14.125	64.75
3206	14.5971	64.75

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

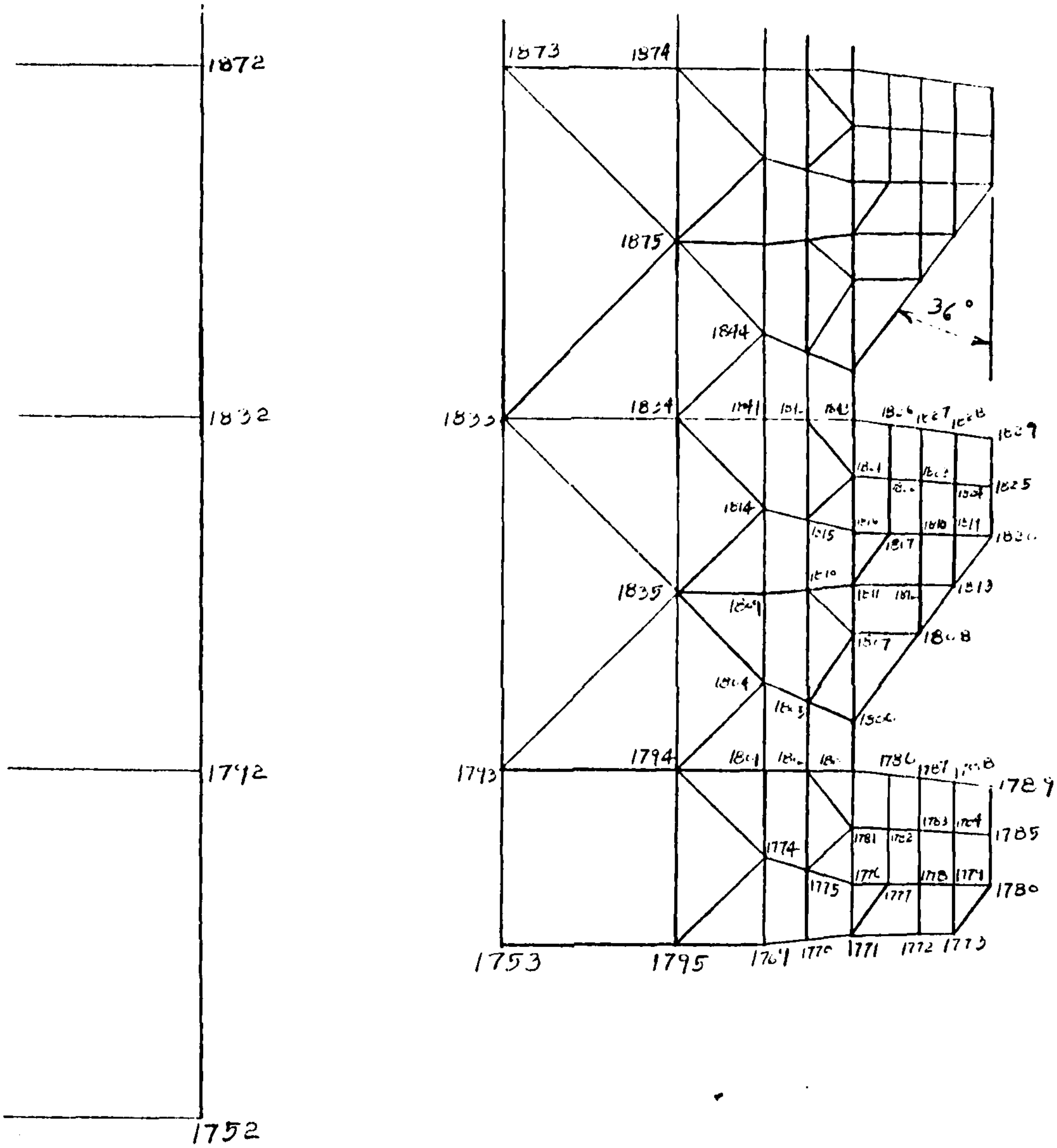
BY DBP DATE 12/22/78 SUBJECT M 14/18 Heater Vessel Bottom End

SHEET NO 1 OF 1

CHKD. BY DATE

PROJ. NO JP1270

Rev. 1 Design

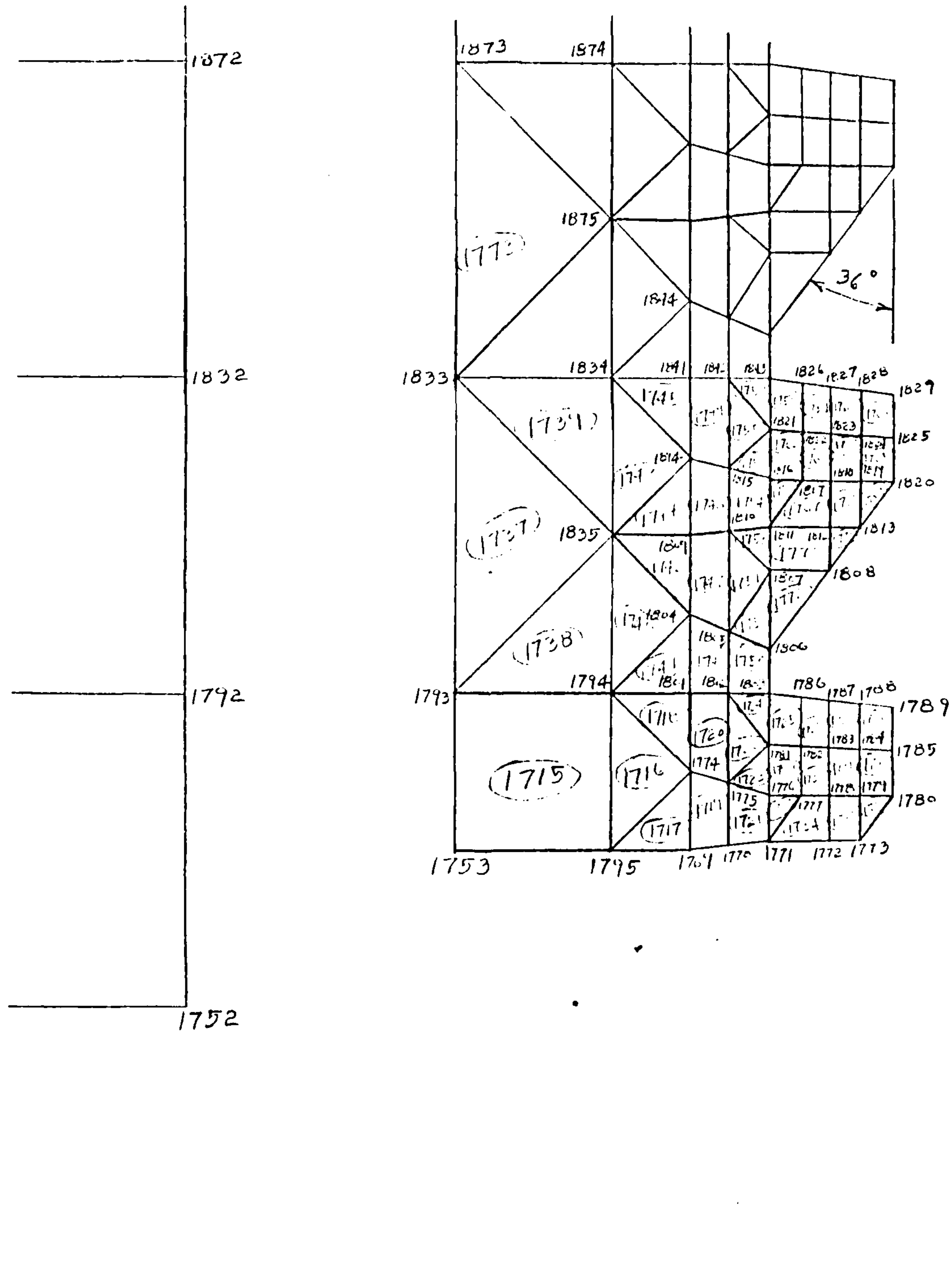


ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 12/22/78 SUBJECT M 14/18 Heater SHEET NO 1 OF 1
CHKD. BY DATE Vessel Bottom End PROJ. NO JP1270

Rev. 1 Design



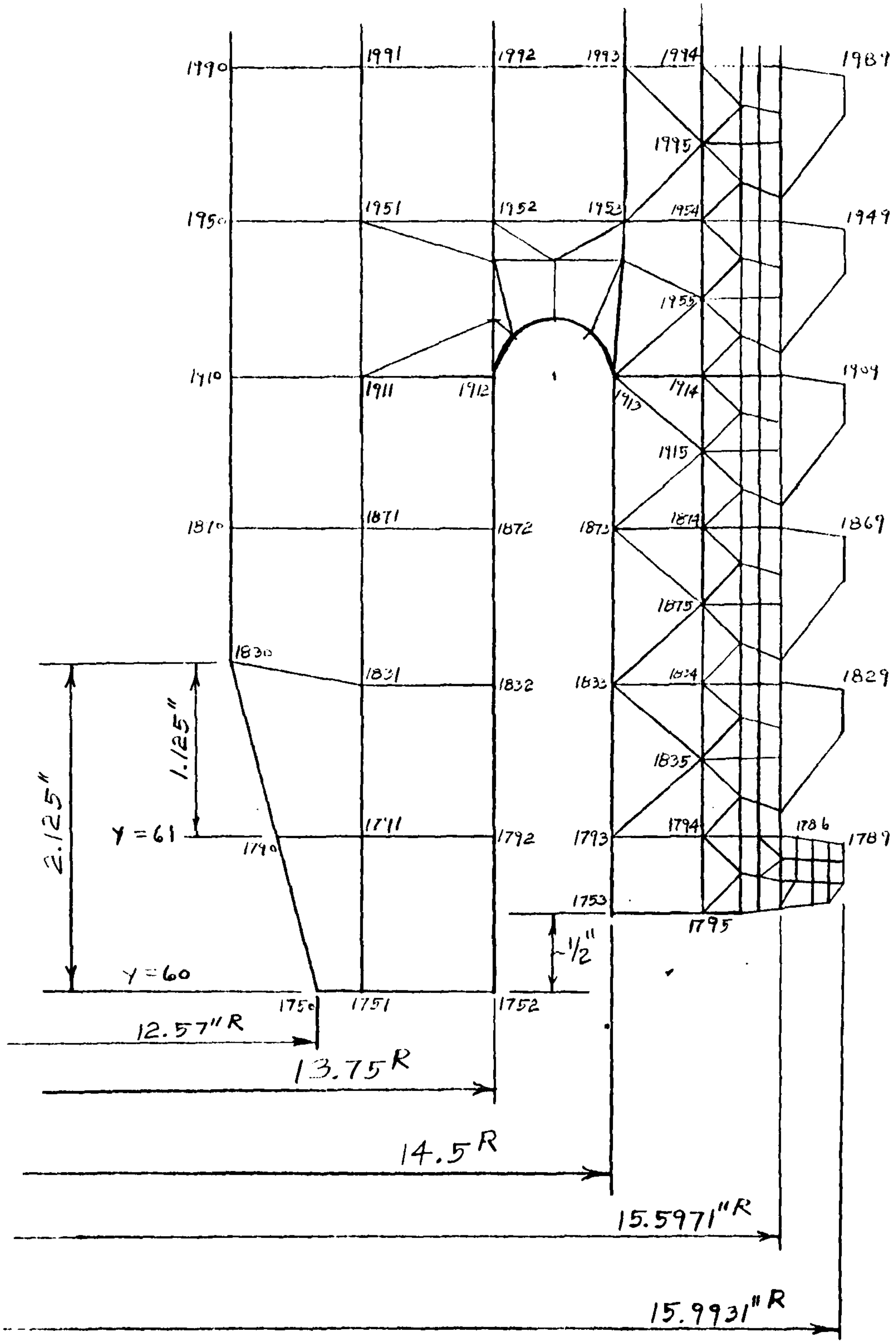
ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 12/22/78 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 1 OF 1

CHKD. BY DATE PROJ. NO JP12

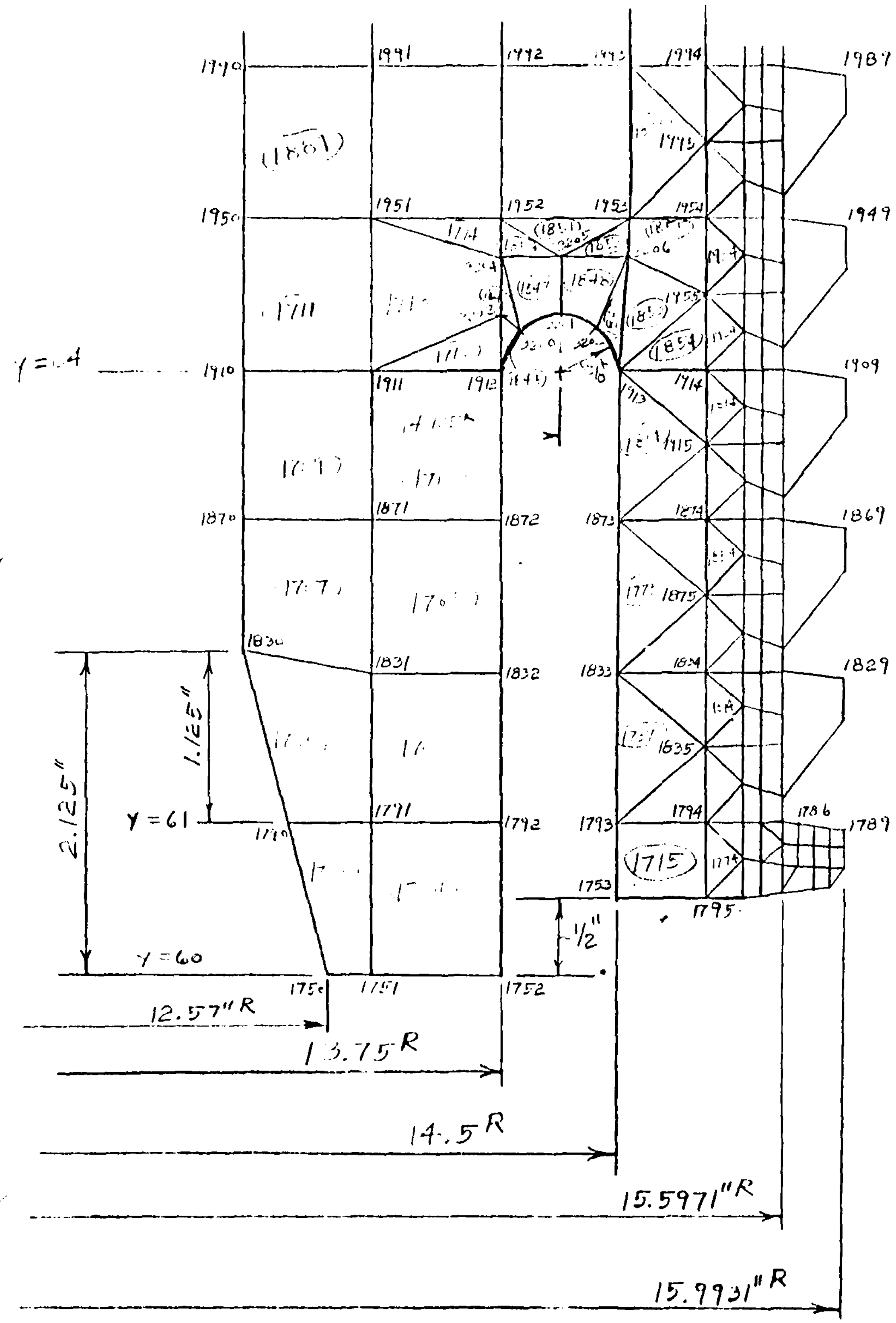
Rev. 1 Design



ENGINEERING DESIGN & ANALYSIS SERVICES — O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 12/22/78 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Rev. 1 Design



ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 1/2/79 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP1270

THREAD LOADS - REV. 1 DESIGN

LOADS in (Lbs/Radian)

THREAD No.	LOAD	THREAD No.	LOAD
1	61,208.4	14	159,787.
2	114,848.4	15	138,784.
3	137,157.	16	118,912.
4	135,186.	17	100,479.
5	152,241.	18	83,651.6
6	232,243.	19	68,494.5
7	270,307.	20	54,987.7
8	270,658.	21	43,059.03
9	258,598.	22	32,598.7
10	242,493.	23	23,487.
11	223,709.	24	15,634.8
12	203,039.	25	9,119.
13	181,435.	26	4,754.

Σ(LOADS) = 3,336,871.13 Lbs/Radian

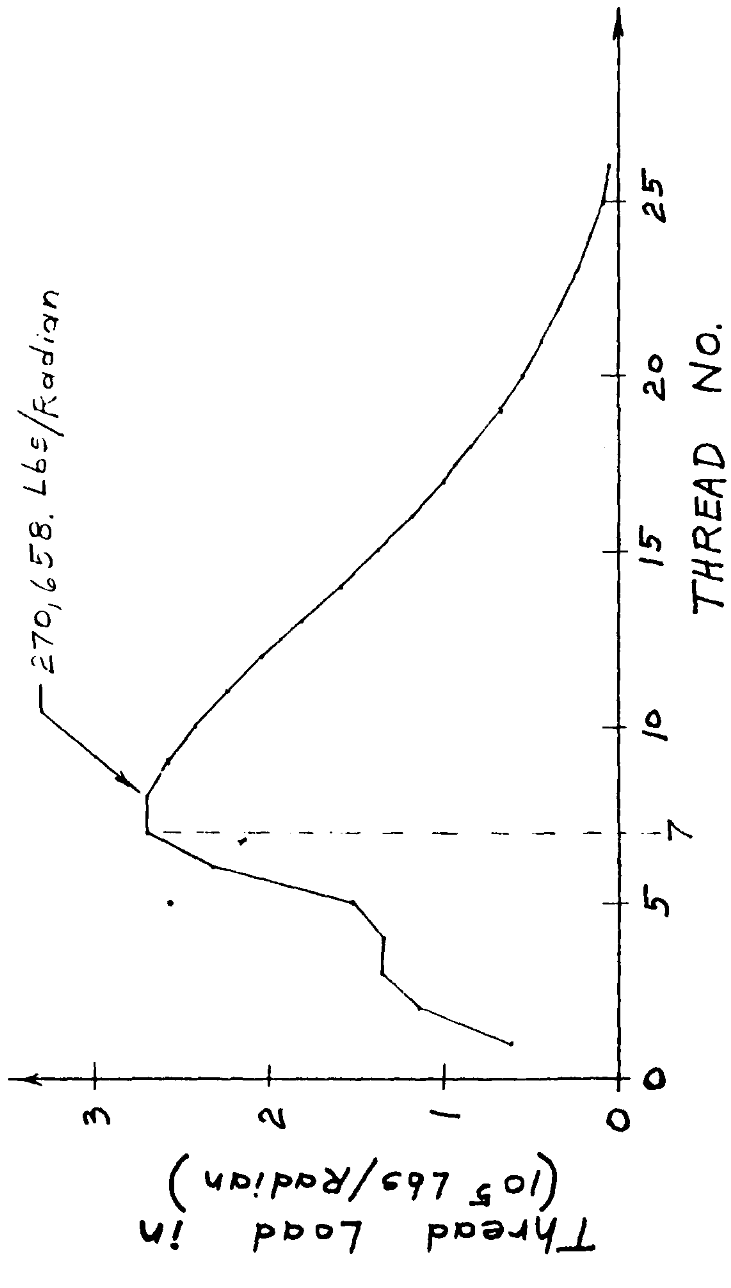
ENGINEERING DESIGN & ANALYSIS SERVICES — O'DONNELL & ASSOCIATES, INC. — PITTSBURGH, PENNSYLVANIA

BY DBP DATE 1/2/79 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 2 OF 2

CHKD. BY DATE PROJ. NO JP/1270

M 14/18 HEATER VESSEL - BOTTOM END

THREAD LOADS FOR REV. 1 DESIGN

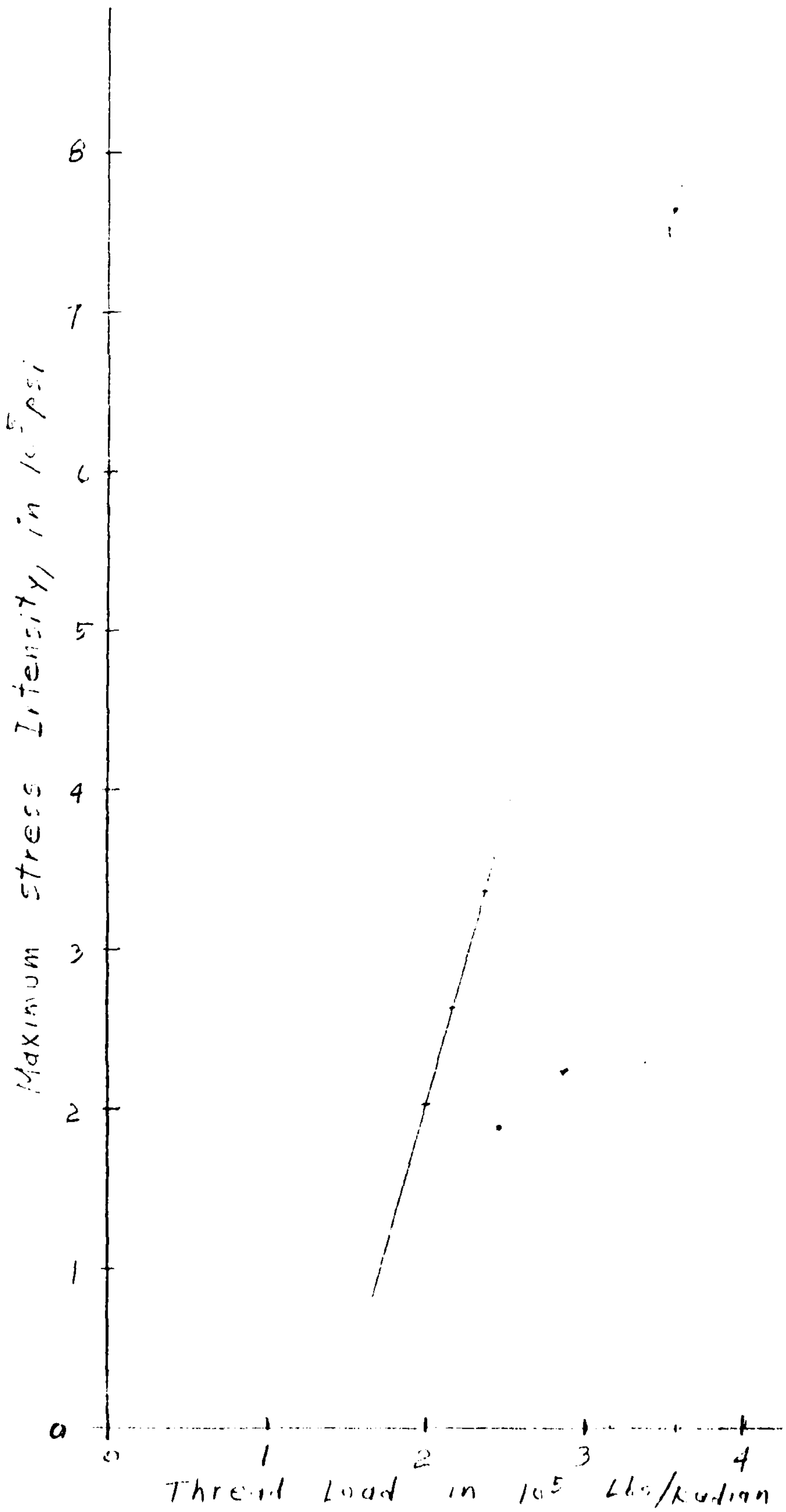


ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 1/5/79 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 1 OF 2

CHKD. BY DATE PROJ. NO JP1270



BY DBP DATE 1/5/79 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 2 OF 2
CHKD. BY DATE PROJ. NO JP1270

Estimated Usage Factor For original Design To DATE

Thread No.	Thread Load (Lbs/Radian)	Max. Stress Intensity (psi)	K	U
1	356,468	765,532	16.642	>1.0
2	352,199	752,000*	16.348	>1.0
3	303,957	580,000*	12.609	0.74
4	265,715	440,000*	9.565	0.37
5	237,652	338,000*	7.348	0.21
6	217,081	263,000*	5.717	0.12
7	200,350	202,242	4.3966	0.01

* Estimated From Thread Loads (see page 1)

BY DBP DATE 1/5/79 SUBJECT M14/18 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Estimated Usage Factor on Original Design To Date

Thread No.	Max. Stress Intensity (psi)	K	U
1	765,532	16.642	>1.0
2	636,933*	13.846	>1.0
3	546,475*	11.88	0.62
4	461,565*	10.034	0.41
5	395,736*	8.603	0.29
6	341,394*	7.422	0.22
7	202,242	4.3966	0.01

* Estimated From Stresses From overall Model

BY DBP DATE 1/5/79 SUBJECT M14/18 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Current
Estimated Usage Factor
For M14/18 Heater Vessel
Bottom End Original Design

Thread No.	Current Usage Factor
1	>1.0
2	>1.0*
3	0.62 → 0.74*
4	0.37 → 0.41*
5	0.21 → 0.29*
6	0.12 → 0.22*
7	0.01

* Estimated

Note: The Usage Factors For Thread No.s 2 Thru 6 were Estimated.

The Usage Factors For Thread No.s 1 and 7 were Calculated From Detail Thread Model Results.

BY DBP DATE 2/2/79 SUBJECT M14 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Equivalent Thread Pressures

DESIGN	THREAD No.	Thread Load (Lbs/Radian)	Thread Pressure (psi)
ORIGINAL	1	356,468	82,412.29
ORIGINAL	2	352,199	81,425.33
ORIGINAL	4	265,715	61,430.99
ORIGINAL	7	200,350	46,319.17
ORIGINAL	10	154,211	35,652.24
REV. 1	1	61,208.4	14,150.85
REV. 1	7	270,307.	62,492.62
REV. 2	1	0	0
REV. 2	2	0	0
REV. 2	4	115,753.2	26,761.13
REV. 2	10	270,066.	62,436.899

$$P = \frac{2(\text{THREAD LOAD}) \cdot \cos(7^{\circ})}{\left[(15.9775)^2 - (15.7065)^2\right]}$$

$$P = 0.23119125\dot{9}(\text{THREAD LOAD})$$

ENGINEERING DESIGN & ANALYSIS SERVICES

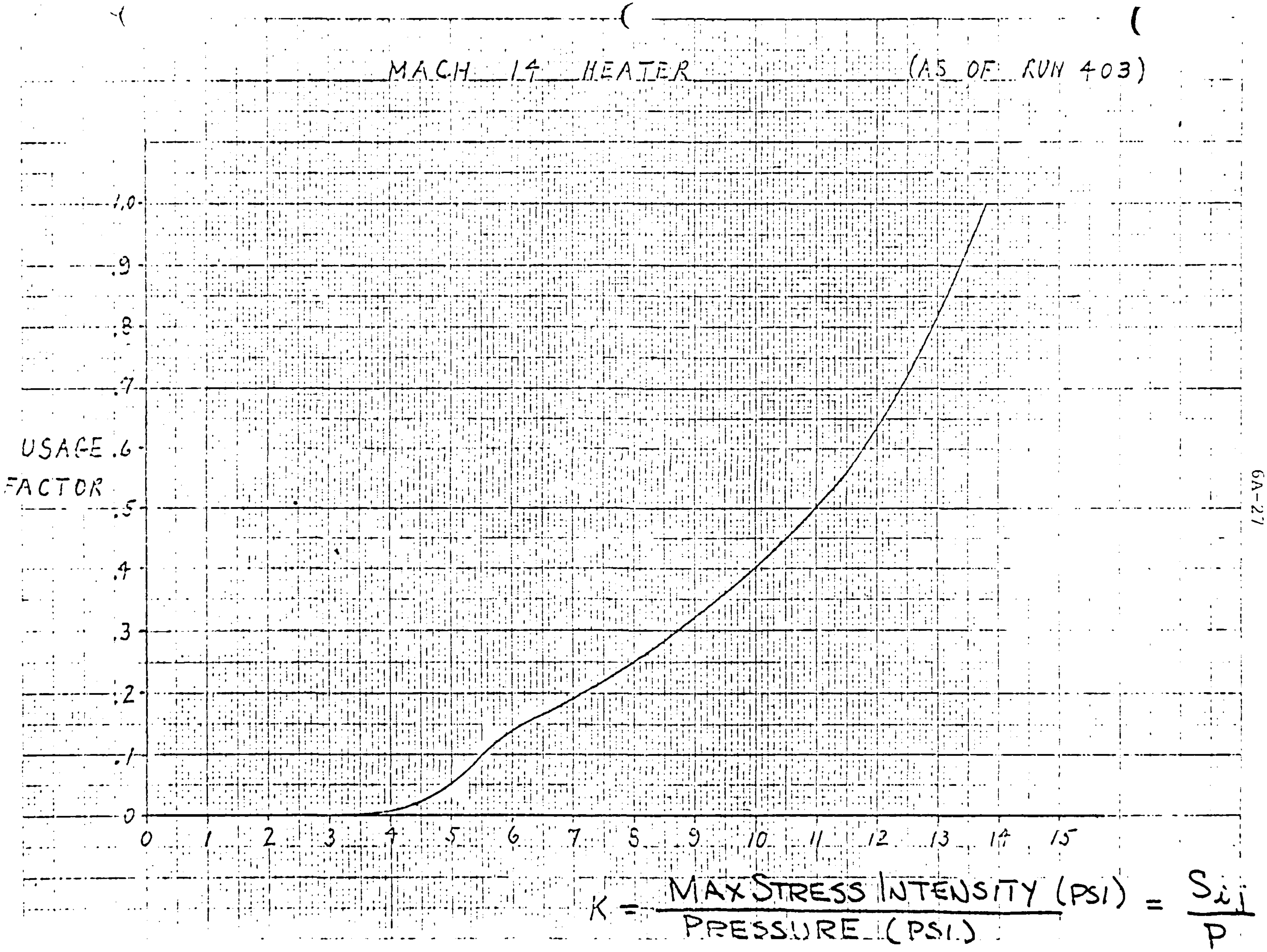
O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 1/4/79 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 1 OF 1

CHKD. BY DATE PROJ. NO JP1270

Interrupted Thread
FACTOR FOR M14/18 Heater
Vessel Bottom End

Pressure (psi)	FACTOR = $\left(\frac{90}{44}\right)\frac{P}{(46,000)}$
46,000	2.0454545
28,000	1.2450593
25,000	1.1116601
20,000	0.8893281
17,000	0.7559289
16,000	0.7114625
15,000	0.6669960
14,000	0.6225296
13,000	0.5780632
10,000	0.4446640
30,000	1.3339921
19,000	0.8448617
18,000	0.8003953
16,500	0.7336957
9,000	0.4001976
8,000	0.3557312

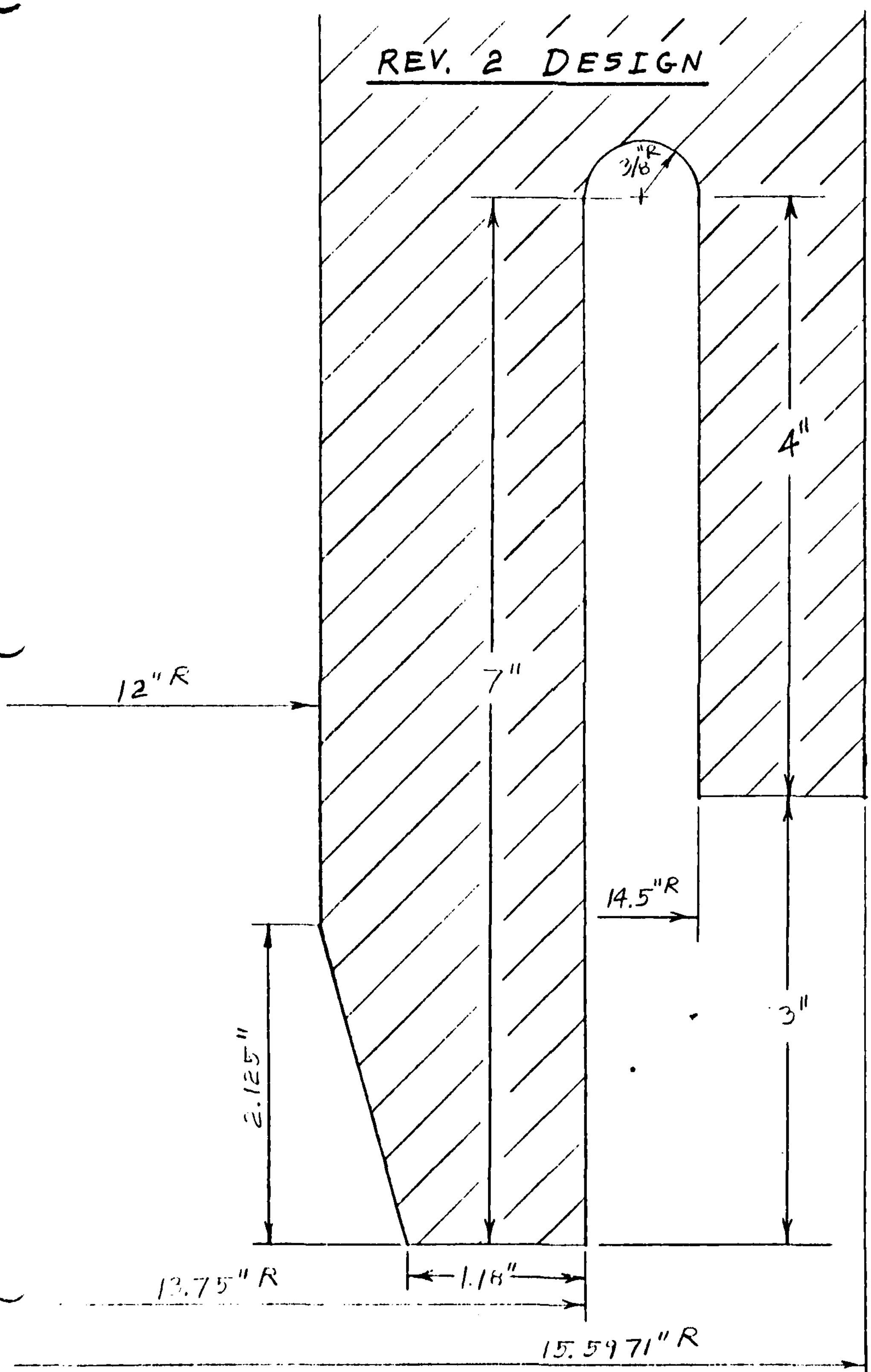

MACH 14 HEATER (AS OF RUN 403)
USAGE FACTOR
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
0
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
$K = \frac{\text{MAX STRESS INTENSITY (PSI)}}{\text{PRESSURE (PSI)}} = \frac{S_{ij}}{P}$

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

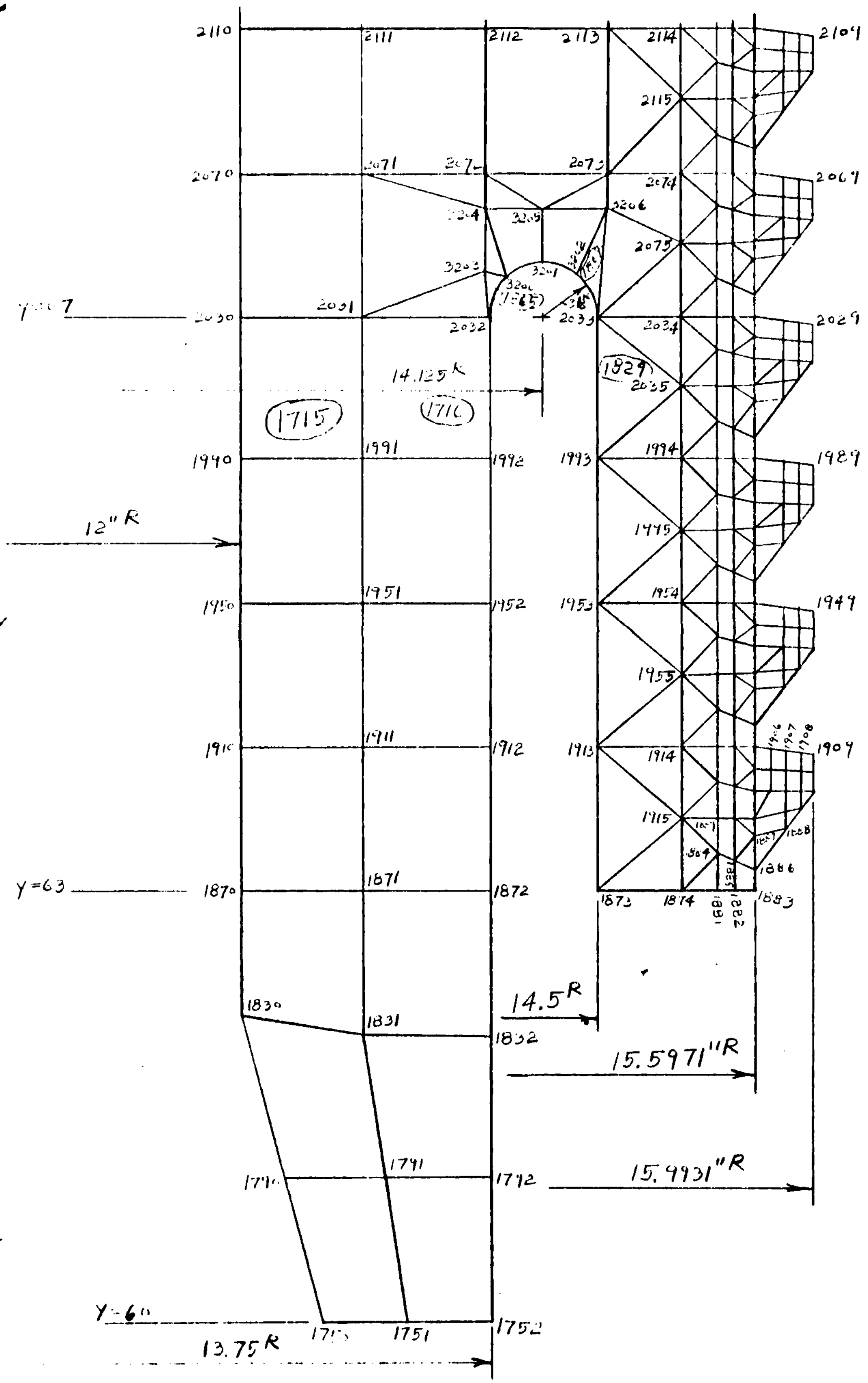
BY DBP DATE 1/5/79 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO. 1 OF 1

CHKD. BY DATE PROJ. NO JP1270



BY DBP DATE 1/5/79 SUBJECT M14/18 HEATER VESSEL BOTTOM End - REV. 2 SHEET NO 1 OF 1

CHKD. BY DATE PROJ. NO JP1270



ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 1/9/79 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 1 OF 2

CHKD. BY DATE PROJ. NO JP1270

THREAD LOADS - REV. 2 DESIGN

LOADS in (Lbs/Radian)

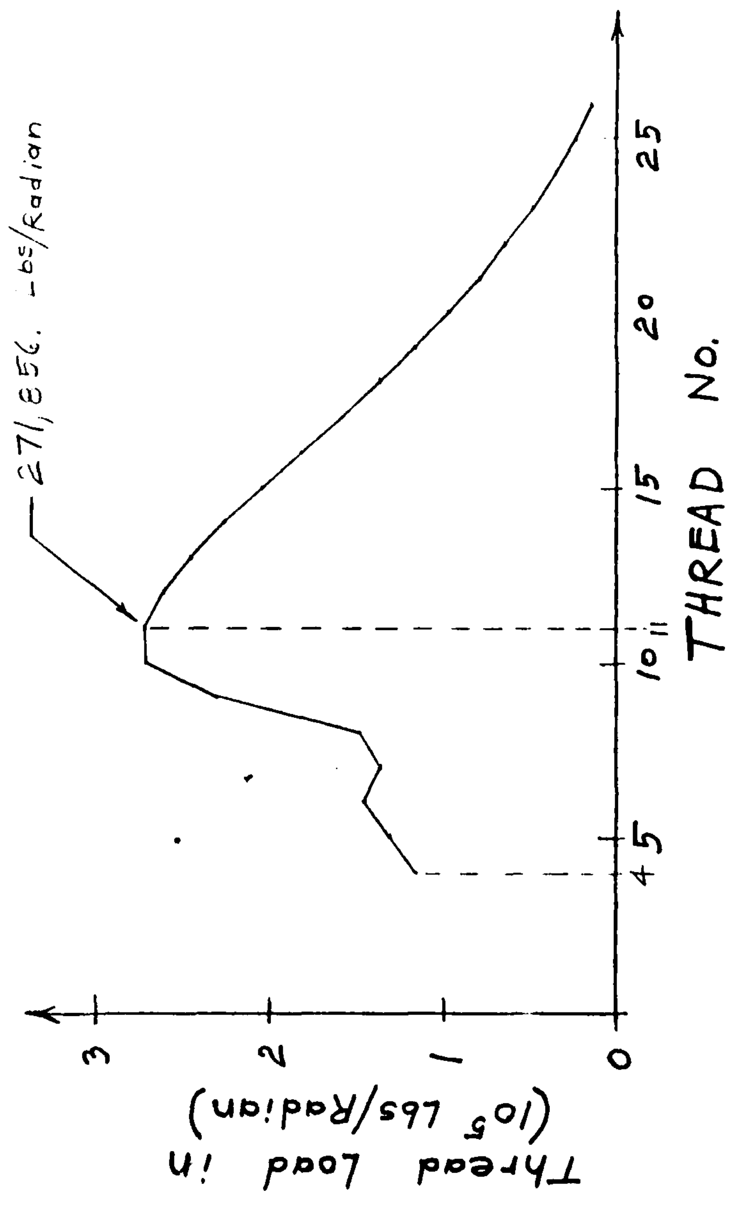
THREAD No.	LOAD	THREAD No.	LOAD
4	115,753.2	16	181,695.
5	130,545.	17	159,290.
6	144,532.	18	137,448.
7	136,269.	19	116,668.
8	148,069.	20	97,263.4
9	229,380.	21	79,407.4
10	270,066.	22	63,154.5
11	271,856.	23	48,474.1
12	260,334.	24	35,287.
13	244,234.	25	23,622.53
14	225,137.	26	14,434.7
15	203,950.		

$\sum(\text{LOADS}) = 3,336,869.83$ Lbs/Radian

BY DBP DATE 1/9/79 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO. 2 OF 2
CHKD. BY DATE PROJ. NO JP1270

M 14/18 HEATER VESSEL - BOTTOM END

THREAD LOADS FOR REV. 2 DESIGN



BY DBP DATE 1/18/79 SUBJECT M14 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

First Thread – Original Design

NODE		DISPLACEMENTS	
Overall Model	Detail Model	δ_X (in)	δ_Y (in)
629	1	-0.202230-3	0.352067-1
	2	-0.353378-3	0.353542-1
	3	-0.504526-3	0.355016-1
	4	-0.655677-3	0.356491-1
630	5	-0.806822-3	0.357965-1
	6	-0.978676-3	0.359882-1
631	7	-0.115053-2	0.361799-1
	21	-0.146217-2	0.366386-1
641	35	-0.177381-2	0.370973-1
	57	-0.182750-2	0.375630-1
693	258	-0.188118-2	0.380286-1
	263	-0.179587-2	0.384632-1
	303	-0.171056-2	0.388977-1
741	306	-0.105550-2	0.417491-1
742	307	-0.973687-3	0.414066-1
743	308	-0.892132-3	0.411679-1
731	309	-0.934528-3	0.406788-1
732	310	-0.118373-2	0.399352-1
	311	-0.143806-2	0.394628-1
733	312	-0.169239-2	0.389903-1

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 1/29/79 SUBJECT M14 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

First Thread – REV. 1 Design

NODE		DISPLACEMENTS	
Overall Model	Detail Model	δ_x (in.)	δ_y (in.)
629	1	-0.645041-3	0.342017-1
	2	-0.782099-3	0.343988-1
	3	-0.919156-3	0.345959-1
	4	-0.105621-2	0.347930-1
630	5	0.119327-2	0.349900-1
	6	-0.137477-2	0.353277-1
631	7	-0.155626-2	0.356653-1
	21	-0.195708-2	0.361416-1
641	35	-0.235790-2	0.366179-1
	57	-0.254531-2	0.370405-1
693	158	-0.273271-2	0.374630-1
	263	-0.285207-2	0.379412-1
	303	-0.297143-2	0.384194-1
741	306	-0.231399-2	0.392212-1
742	307	-0.229514-2	0.390805-1
743	308	-0.226767-2	0.390434-1
731	309	-0.233025-2	0.389284-1
732	310	-0.255255-2	0.387352-1
	311	-0.277471-2	0.386283-1
733	312	-0.299686-2	0.385213-1

ENGINEERING DESIGN & ANALYSIS SERVICES

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PITTSBURGH, PENNSYLVANIA

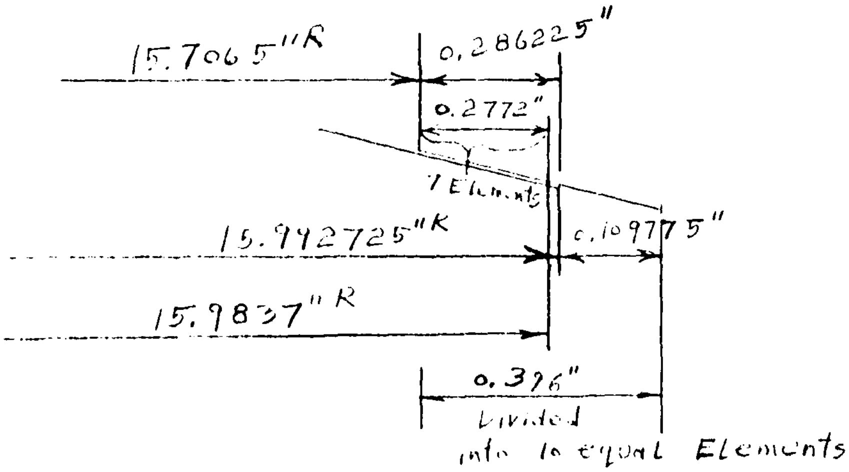
BY DBP DATE 1/29/79 SUBJECT M14 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

First Thread – REV. 2 Design

NODE		DISPLACEMENTS	
Overall Model	Detail Model	δ_x (in.)	δ_y (in.)
629	1	-0.843166-3	0.335081-1
	2	-0.970933-3	0.337357-1
	3	-0.109870-2	0.339632-1
	4	-0.122647-2	0.341908-1
630	5	-0.135423-2	0.344183-1
	6	-0.153444-2	0.348397-1
631	7	-0.171464-2	0.352610-1
	21	-0.216838-2	0.357322-1
641	35	-0.262212-2	0.362034-1
	57	-0.288088-2	0.365798-1
693	258	-0.313964-2	0.369562-1
	263	-0.336631-2	0.374444-1
	303	-0.359298-2	0.379327-1
741	306	-0.282869-2	0.377122-1
742	307	-0.286267-2	0.376788-1
743	308	-0.288377-2	0.377304-1
751	309	-0.296873-2	0.377821-1
752	310	-0.318722-2	0.378550-1
	311	-0.341424-2	0.379459-1
753	312	-0.364126-2	0.380367-1

BY DBP DATE 1/18/79 SUBJECT M14 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Pressure on 1st Thread - original Design



Equivalent Thread Pressure

Load = 356,468 Lbs/Radian

$$P = \frac{2(356,468) \cdot \cos(7^\circ)}{\left[(15.9837)^2 - (15.7065)^2\right]} = 80,553.25 \text{ psi}$$

$P = 0.225976095$ (Thread Load)

For 1st Thread - REV. 1 Design

Thread Load = 61,208.4

P = 13,831.64 psi

For 2nd Thread - REV. 2 Design

Thread Load = P = 0

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PITTSBURGH, PENNSYLVANIA

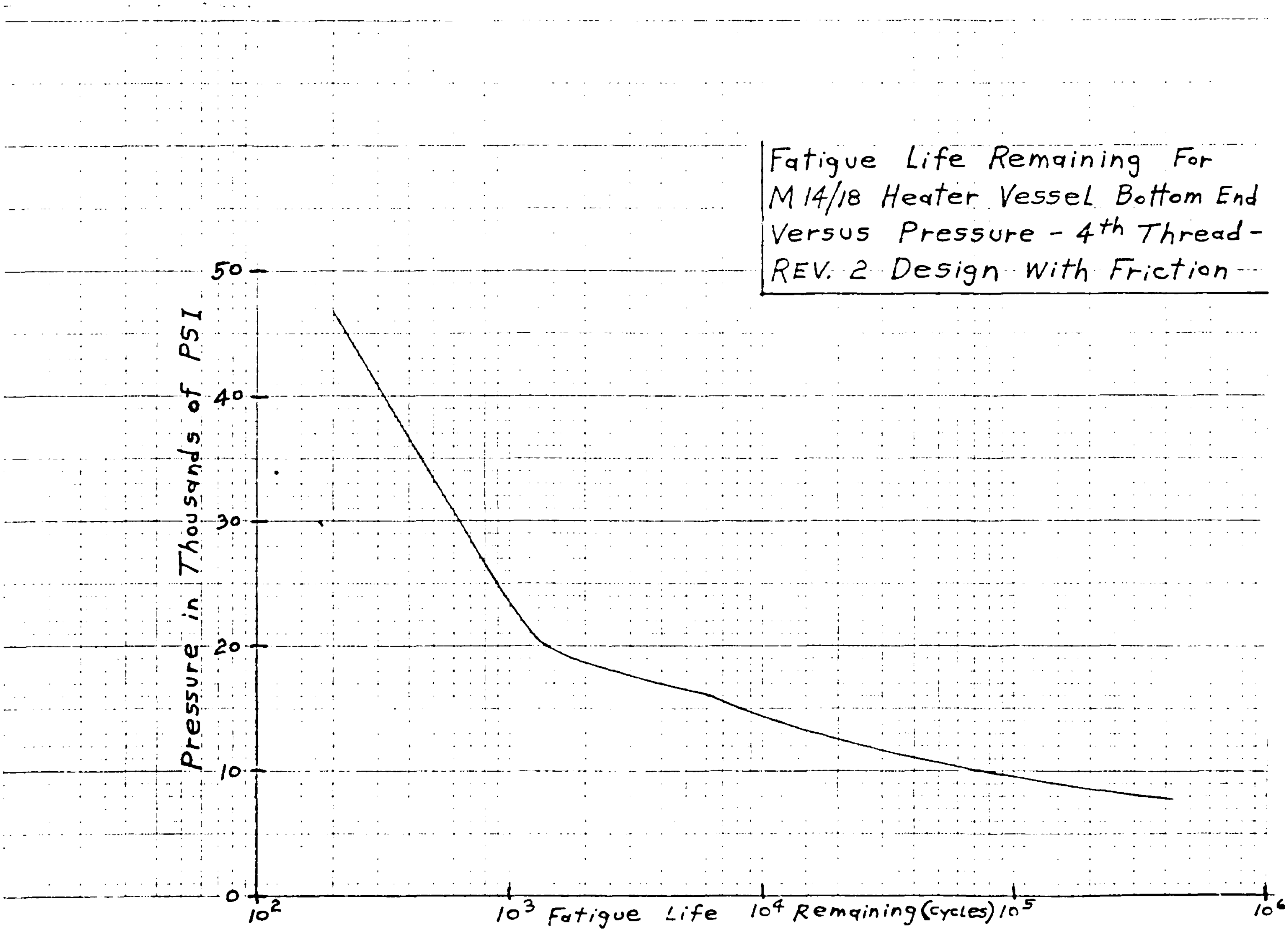
BY DBP DATE 2/8/79 SUBJECT M14/18 Heater Vessel Bottom End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Fatigue Life of M14/18 Heater Vessel Bottom End vs. P – 4th Thread – REV. 2 Design – with Friction

P (psi)	Fatigue Life (cycles)	Fatigue Life Remaining (cycles)
46,000	256	210
30,000	766	628
28,000	879	721
25,000	1,096	899
20,000	1,688	1,384
17,000	4,832	3,962
16,000	7,414	6,079
15,000	10,012	8,210
14,000	13,986	11,469
13,000	20,326	16,667
10,000	88,232	72,350
19,000	2,134	1,750
18,000	3,157	2,589
16,500	6,069	4,977
9,000	172,067	141,095
8,000	385,462	316,079

N_R = Fatigue Life Remaining

$$N_R = 0.82\,(\text{Fatigue Life})$$


Fatigue Life Remaining For
M 14/18 Heater Vessel Bottom End
Versus Pressure - 4th Thread -
REV. 2 Design With Friction
Pressure in Thousands of PSI
50
40
30
20
10
0
10^2
10^3 Fatigue Life 10^4 Remaining (cycles) 10^5
10^6

BY DBP DATE 2/12/79 SUBJECT M 14/18 Heater Vessel Bottom End SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP1270

With Friction

Bottom End - 4th Thread - REV. 2 Design - P=28,000 psi

If $\sigma = \Delta\sigma = 186{,}658$ psi and $K_{IC} = 100$ ksi$\sqrt{in}$

1. $K_{IC} = 100$ ksi$\sqrt{in}$

2. Critical Crack Depth

$$a_{cr} = \frac{1}{1.25\,\pi}\left(\frac{100{,}000}{186{,}658}\right)^2 = 0.073088''$$

3. Cycles to Failure

$C_o = 1.17366 \times 10^{-15}$ for ΔK in psi$\sqrt{in}$

$(n-2) = 0.25$ $\quad M^{n/2} = (1.25\,\pi)^{1.125} = 4.659264564$

$\Delta\sigma^n = (186{,}658)^{2.25} = 7.241939099 \times 10^{11}$

$$\frac{1}{a_{cr}^{(n-2)/2}} = \frac{1}{(0.073088)^{0.125}} = 1.386816939$$

$$N = 2{,}020.121815\left[\frac{1}{a_i^{0.125}} - 1.386816939\right]$$

$$a_i = \left(\frac{2{,}020.121815}{N + 2{,}801.537637}\right)^8$$

ENGINEERING DESIGN & ANALYSIS SERVICES — O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

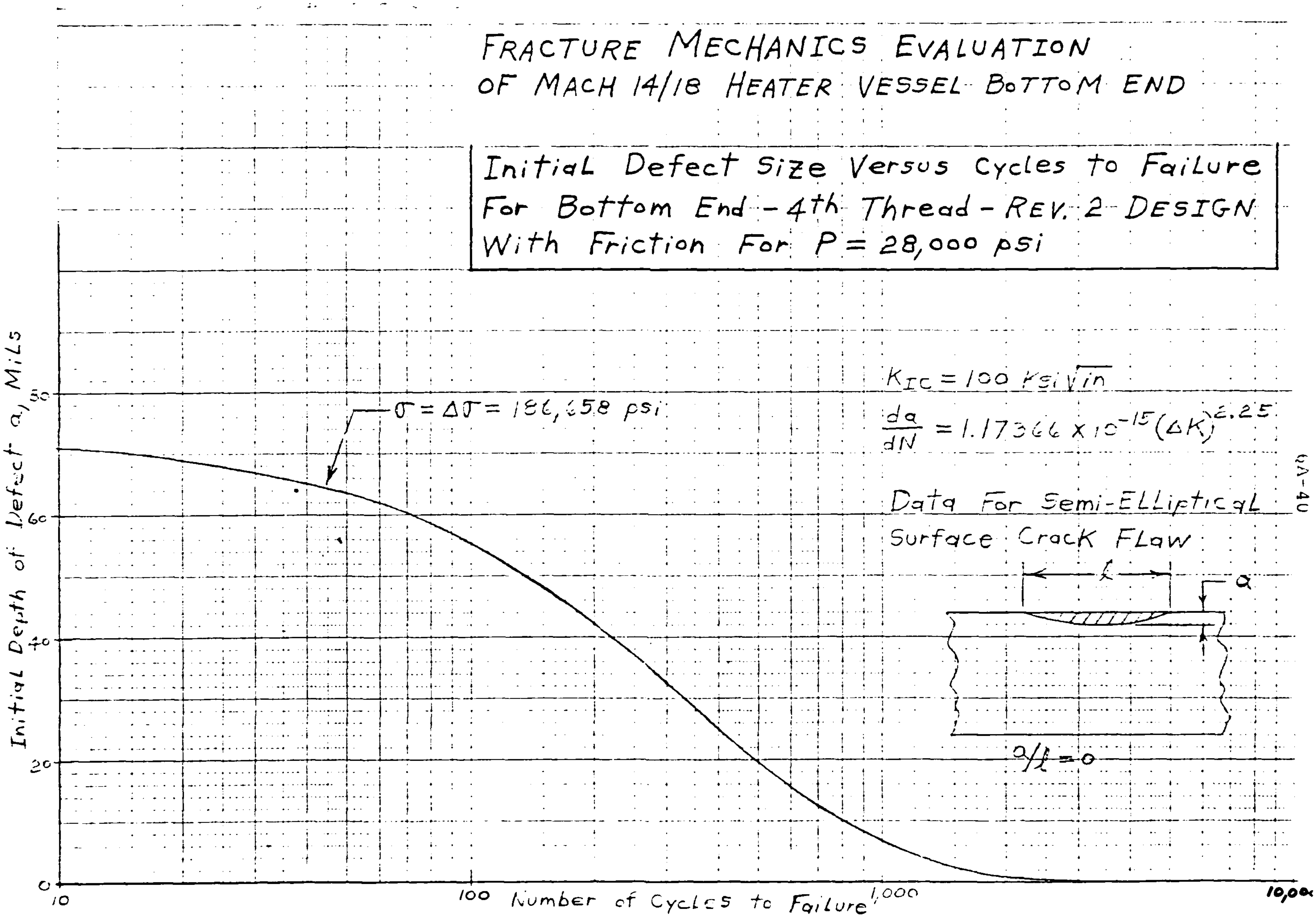
BY DBP DATE 2/12/79 SUBJECT M14/18 Heater Vessel Bottom End SHEET NO 2 OF 2
CHKD. BY DATE PROJ. NO JP1270

M14/18 Heater Vessel Bottom End – 4th Thread – Rev. 2 Design – with Friction – P = 28,000 psi

a_i Versus N for Threads on Bottom End Closure
$\sigma = \Delta\sigma = 186,658$ psi, $K_{IC} = 100\ ksi\sqrt{in}$
Modified AISI 4340 Material

a_i inches	N Cycles
0.07103449	10
0.06904523	20
0.06344357	50
0.05520708	100
0.04209884	200
0.03238948	300
0.01964674	500
0.01227298	700
0.006358375	1,000
0.0009816977	2,000
0.0002161103	3,000
0.0000605568	4,000
0.00002021058	5,000
0.0000077701	6,000

$$a_i = \left(\frac{2,020.121815}{N + 2,801.537637}\right)^8$$


FRACTURE MECHANICS EVALUATION
OF MACH 14/18 HEATER VESSEL BOTTOM END
Initial Defect Size Versus Cycles to Failure
For Bottom End - 4th Thread - REV. 2 DESIGN
With Friction For P = 28,000 psi
σ = Δσ = 186,658 psi
$K_{IC} = 100$ ksi$\sqrt{in}$
$\frac{da}{dN} = 1.17366 \times 10^{-15} (\Delta K)^{2.25}$
Data For Semi-Elliptical
Surface Crack Flaw
ℓ
a
a/ℓ = 0
Initial Depth of Defect a, Mils
0
20
40
60
80
10
100
1,000
10,000
Number of Cycles to Failure

APPENDIX 7A

DESIGN MODIFICATIONS
TO DRIVER VESSEL

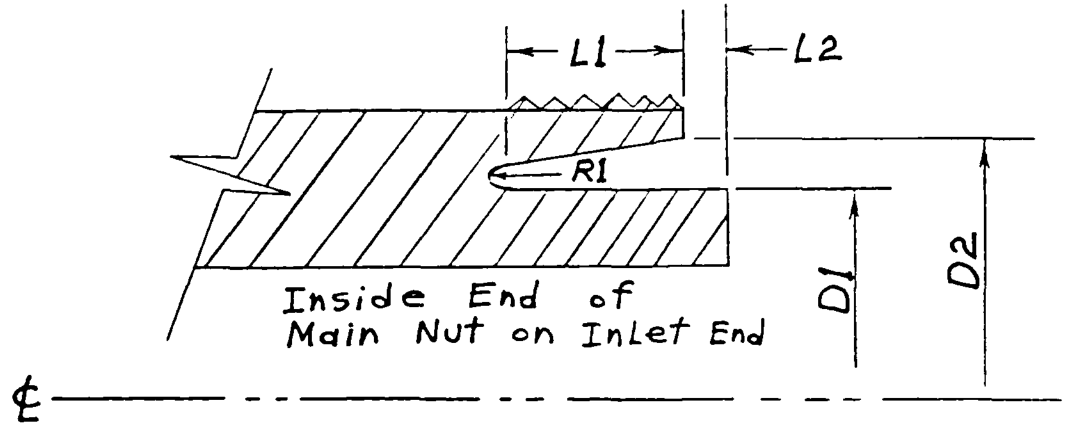
BY DBP   DATE 12/15/78   SUBJECT DRIVER VESSEL INLET END   SHEET NO. 1 OF 1
CHKD BY   DATE   PROJ. NO JP1270

Summary of Fatigue Life Remaining on Modified Driver Vessel Inlet End Based on P = 47,500 psi

DESIGN	L1 (inches)	L2 (inches)	D1 (inches)	D2 (inches)	Critical Thread No.	Life Remaining No Friction	Life Remaining with Friction
Original	0	0	—	—	2	222 cycles	152 cycles
REV. 1*	4	1/2	32	35 1/4	7	423 cycles	—
REV. 2*	5	1/2	32	35 1/4	8	447 cycles	—
REV. 3*	5	1/2	32	33 1/2	8	495 cycles	—
REV. 4*	4	1/2	31 1/2	33	8	515 cycles	389 cycles

*R1 = 3/8"

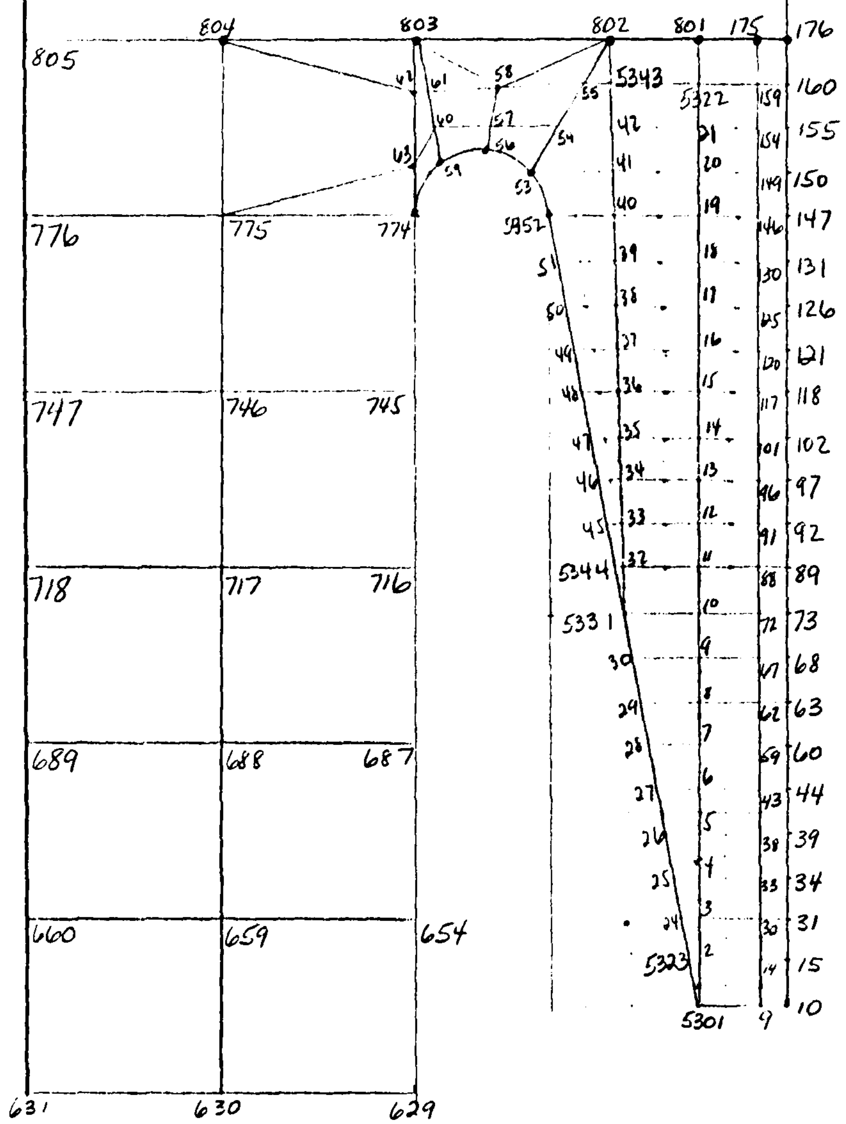
Note: With Friction, A coefficient of Friction, f, of f = 0.12278 was Used.



BY DATE SUBJECT SHEET NO OF

CHKD BY DATE PROJ. NO

REV 2 DESIGN



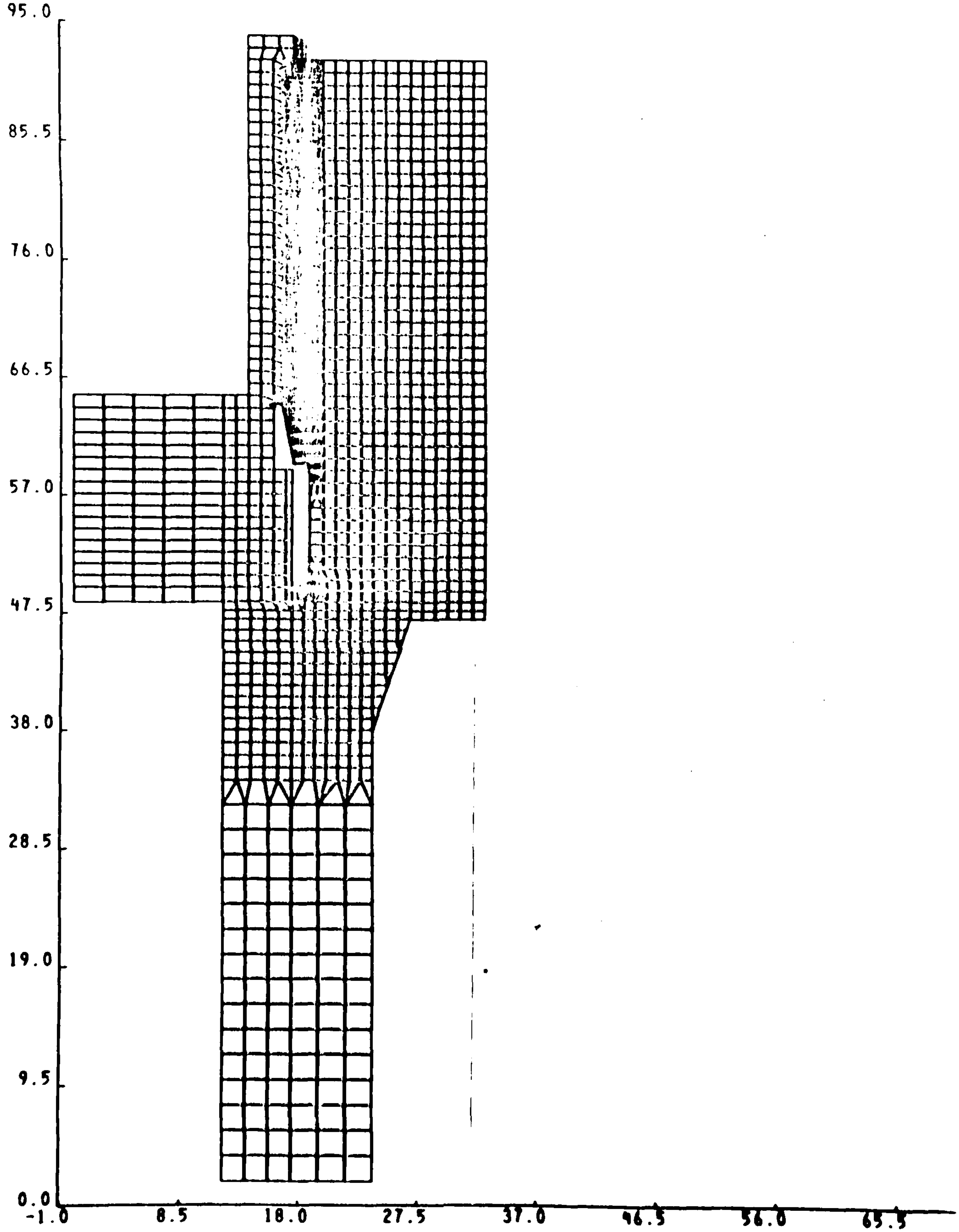
16.375, 64.062

$y = mx + b$

$x = \frac{y - b}{m}$

$m = -5.14285$

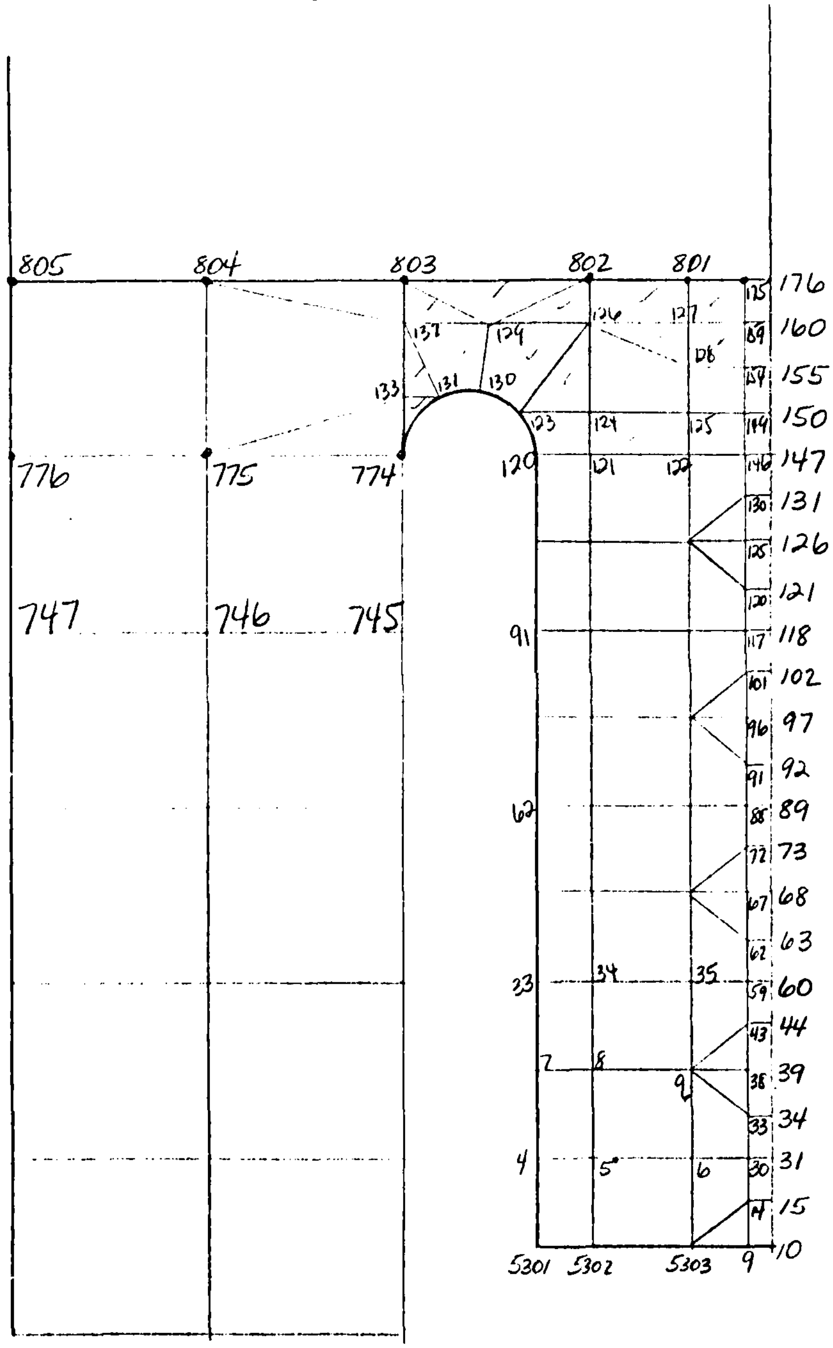
150.0048

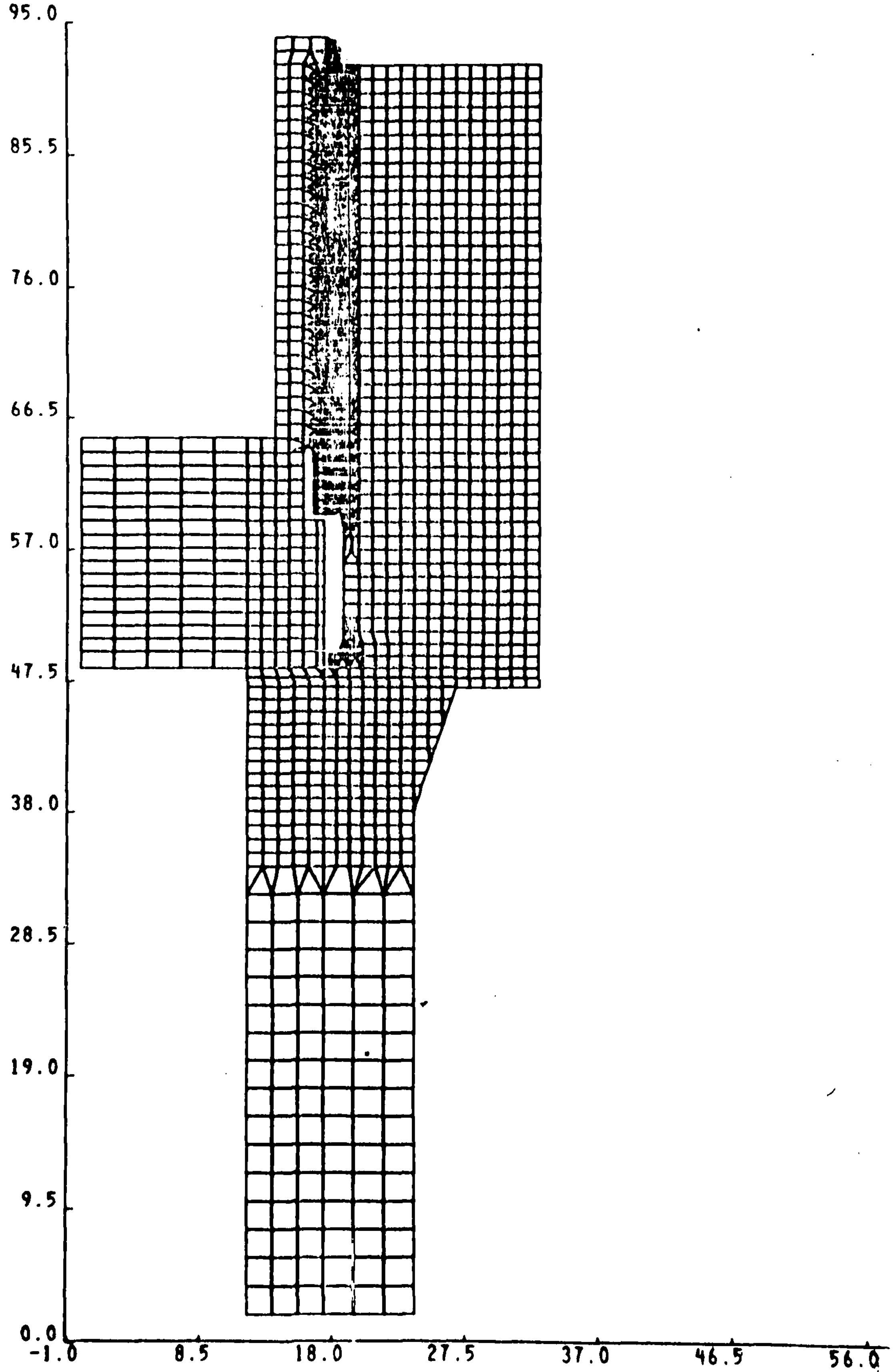


INLET END OF MACH 10 DRIVER VESSEL - REV. 2 DESIGN - THREAD ANGLE CORR.

BY DATE SUBJECT SHEET NO OF
CHKD. BY DATE PROJ. NO

REV 3 DESIGN





INLET END OF MACH 10 DRIVER VESSEL - REV. 3 DESIGN - THREAD ANGLE CORR

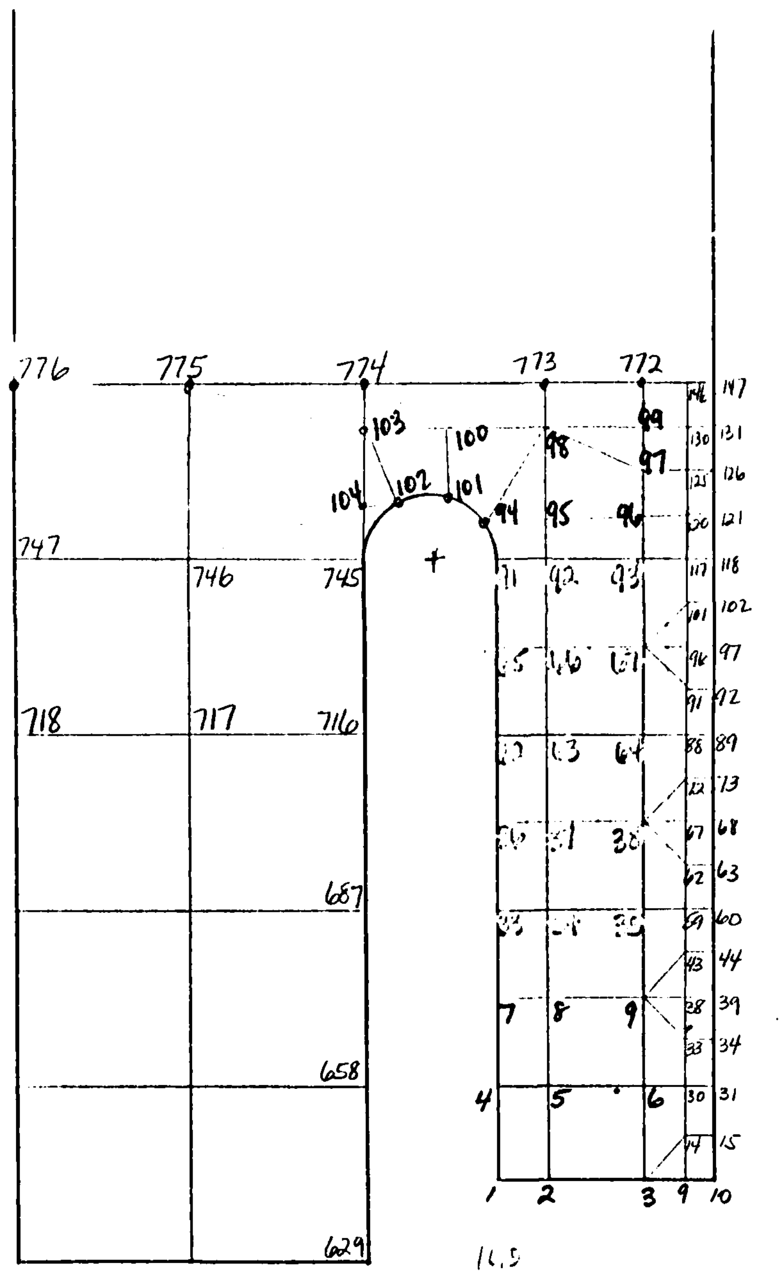
ENGINEERING DESIGN & ANALYSIS SERVICES 7A-6

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY ____ DATE ____ SUBJECT ____ SHEET NO ____ OF ____

CHKD. BY ____ DATE ____ PROJ. NO ____

REV. 4 DESIGN



ϕ31.5

12.915

ENGINEERING DESIGN & ANALYSIS SERVICES — O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY ELW DATE 11/23/78 SUBJECT Gas storage Vessel Inlet End SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP1270

THREAD LOADS - REV. 1 DESIGN

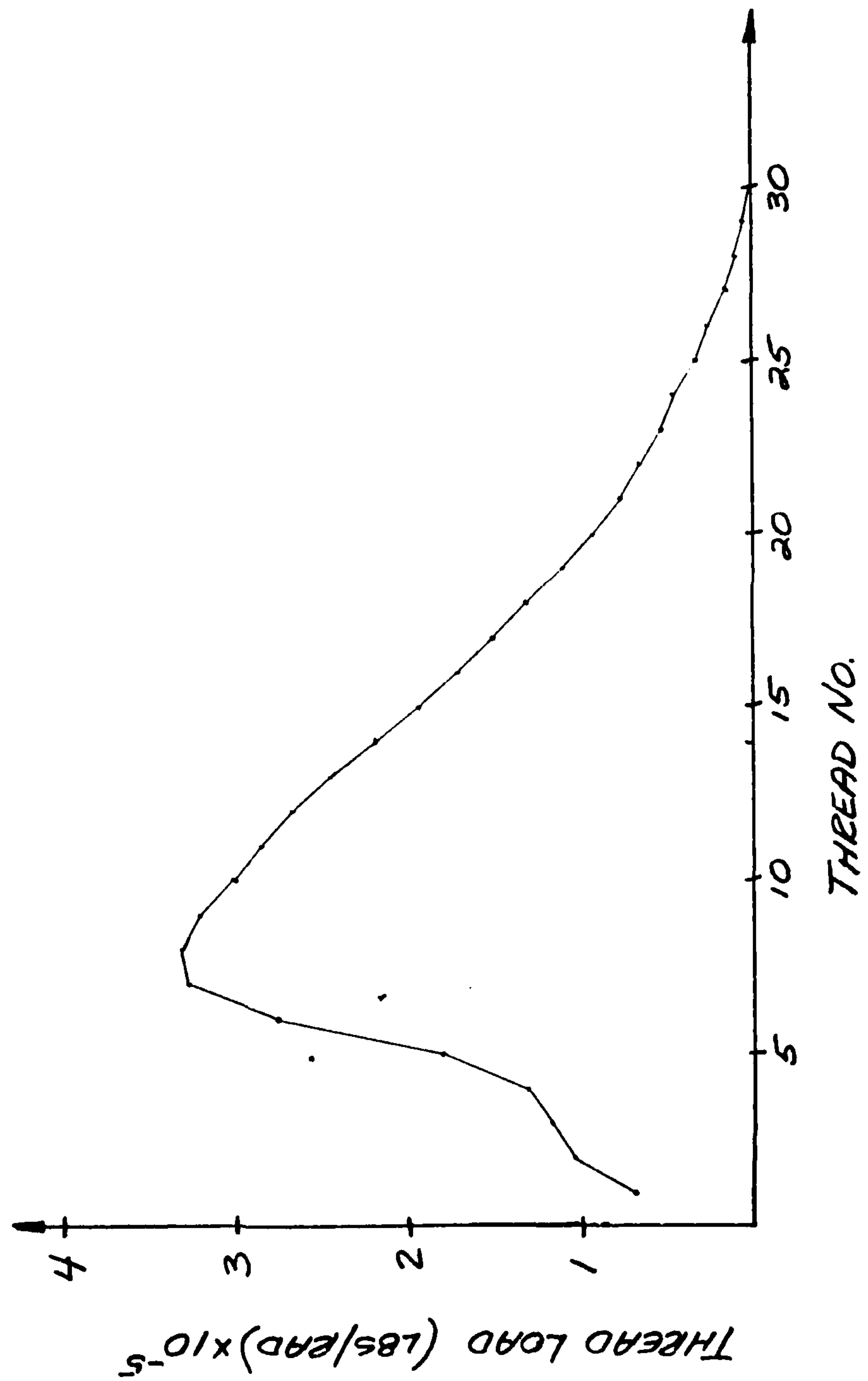
LOADS IN $(LBS/RAD) \times 10^{-5}$

THREAD NO.	LOAD	THREAD NO.	LOAD
1	.695	17	1.509
2	1.014	18	1.307
3	1.191	19	1.120
4	1.333	20	0.950
5	1.805	21	0.796
6	2.795	22	0.658
7	3.266	23	0.535
8	3.303	24	0.427
9	3.201	25	0.332
10	3.044	26	0.249
11	2.856	27	0.176
12	2.645	28	0.111
13	2.418	29	0.054
14	2.185	30	0.002
15	1.952	31	0.0
16	1.725	32	0.0

BY ELW DATE 11/23/78 SUBJECT Gas storage Vessel Inlet End SHEET NO 2 OF 2
CHKD. BY DATE PROJ. NO JP1270

DRIVER VESSEL - INLET END - THREAD LOADS

REV. 1 - DESIGN



ENGINEERING DESIGN & ANALYSIS SERVICES | O'DONNELL & ASSOCIATES, INC. PITTSBURGH, PENNSYLVANIA

BY DBP DATE 11/30/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP/270

THREAD LOADS - REV. 2 DESIGN

LOADS in (Lbs/Radian)

THREAD No.	LOAD	THREAD No.	LOAD
1	68,157.	17	164,998.
2	102,971.8	18	143,892.
3	120,048.6	19	124,182.
4	129,012.62	20	106,020.
5	134,258.6	21	89,476.
6	173,972.	22	74,550.2
7	277,487.	23	61,197.
8	326,079.	24	49,330.3
9	327,547.	25	38,839.1
10	314,969.	26	29,588.64
11	297,740.	27	21,426.
12	277,941.	28	14,181.3
13	256,287.	29	7,679.05
14	233,479.	30	1,756.4
15	210,247.	31	-3,680.5
16	187,246.	32	-8,432.57

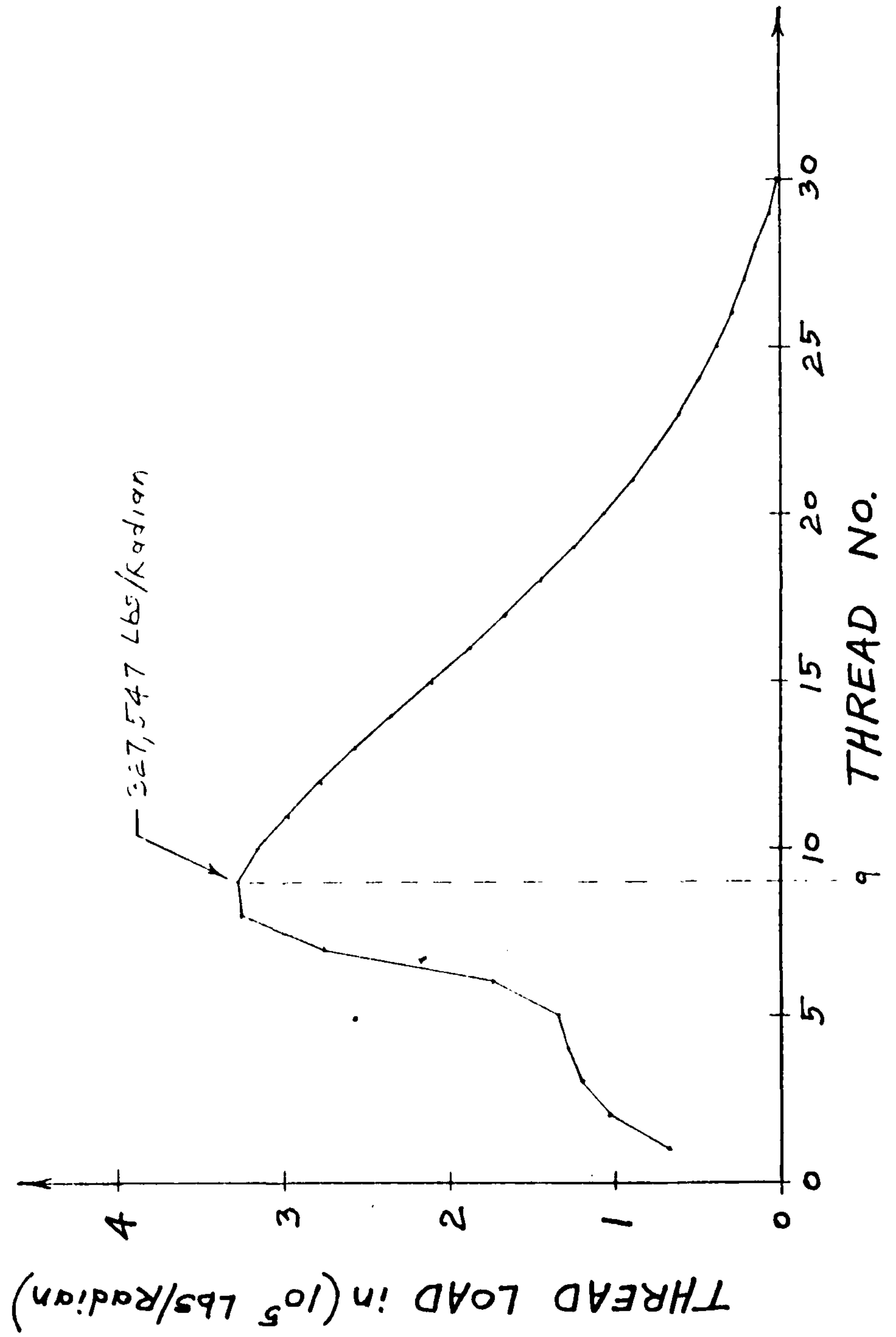
Σ (LOADS) = 4,352,445.54 LBS/Radian

ENGINEERING DESIGN & ANALYSIS SERVICES — O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 11/30/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 2 OF 2
CHKD. BY DATE PROJ. NO JP1270

DRIVER VESSEL - INLET END - THREAD LOADS

REV. 2 - DESIGN



BY DBP DATE 11/30/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP1270

THREAD LOADS - REV. 3 DESIGN

LOADS in (Lbs/Radian)

THREAD No.	LOAD	THREAD No.	LOAD
1	85,670.	17	159,353.
2	130,114.1	18	138,564.
3	156,333.	19	119,205.
4	169,440.	20	101,420.
5	160,105.	21	85,268.1
6	167,363.	22	70,741.4
7	260,081.	23	57,784.3
8	310,535.	24	46,307.7
9	315,672.	25	36,193.1
10	305,789.	26	27,303.5
11	290,065.	27	19,484.
12	271,003.	28	12,567.16
13	249,713.	29	6,376.13
14	227,129.	30	751.1
15	204,098.	31	-4,402.3
16	181,326.	32	-8,905.86

Σ(Loads) = 4,352,446.43 Lbs/Radian

ENGINEERING DESIGN & ANALYSIS SERVICES

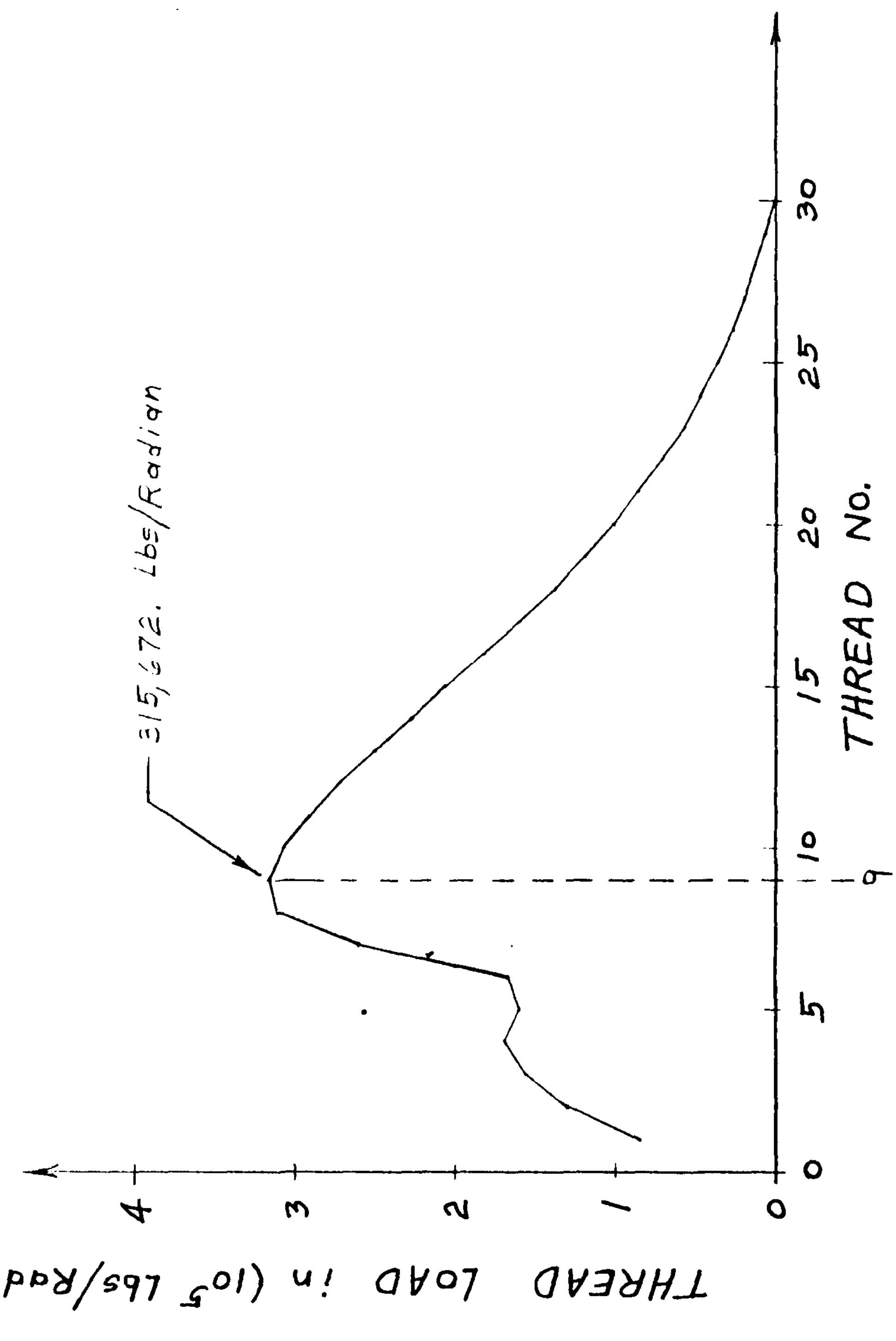
O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 11/30/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 2 OF 2

CHKD. BY DATE PROJ. NO JP1270

DRIVER VESSEL - INLET END - THREAD LOADS

REV. 3 DESIGN



ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 12/7/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP1270

THREAD LOADS - REV. 4 DESIGN

LOADS in (Lbs/Radian)

THREAD No.	LOAD	THREAD No.	LOAD
1	85,503.	17	151,947.
2	132,189.05	18	131,131.
3	155,578.6	19	111,929.
4	158,289.	20	94,444.
5	168,553.	21	78,694.9
6	239,114.1	22	64,646.
7	294,415.	23	52,213.5
8	313,093.	24	41,286.4
9	313,492.	25	31,730.7
10	303,841.	26	23,398.07
11	287,812.	27	16,124.8
12	267,655.	28	9,738.28
13	245,068.	29	4,063.3
14	221,349.	30	-1,057.
15	197,479.	31	-5,711.84
16	174,175.	32	-9,740.76

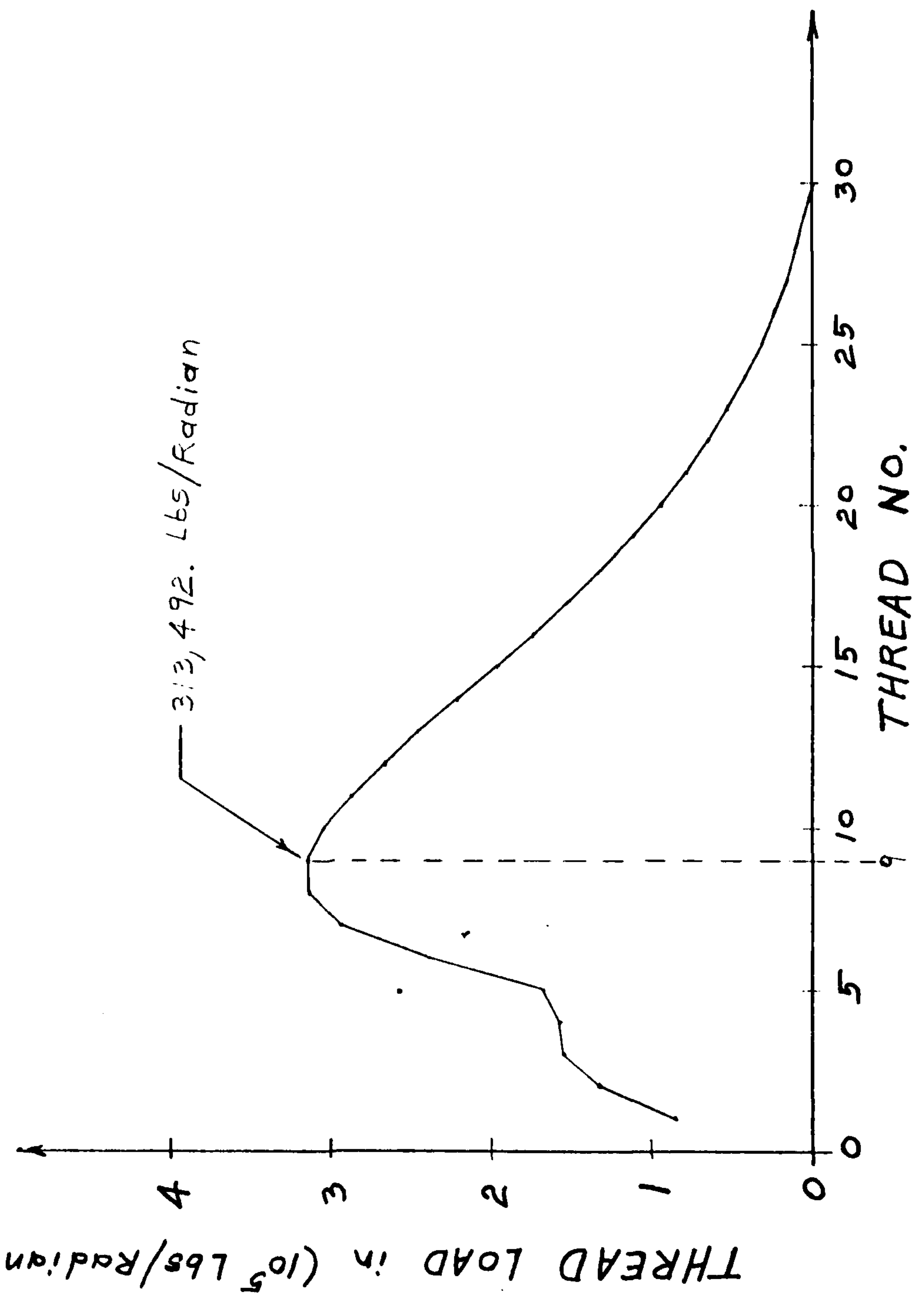
Σ(LOADS) = 4,352,443.1 Lbs/Radian

ENGINEERING DESIGN & ANALYSIS SERVICES — O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 12/7/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 2 OF 2
CHKD. BY DATE PROJ. NO JP1270

DRIVER VESSEL - INLET END - THREAD LOADS

REV. 4 DESIGN



ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 1/30/79 SUBJECT Driver Vessel Inlet End SHEET NO 1 OF 1

CHKD. BY DATE PROJ. NO JP1270

First Thread- Original Design

NODE		DISPLACEMENTS	
Overall Model	Detail Model	δ_x (in)	δ_y (in)
1727	1	-0.377202-2	0.326231-1
	2	-0.400194-2	0.328434-1
1729	3	-0.423186-2	0.330636-1
	4	-0.435833-2	0.331732-1
1730	5	-0.448479-2	0.332828-1
	6	-0.465564-2	0.335358-1
1731	7	-0.482648-2	0.337887-1
	21	-0.515230-2	0.343403-1
1749	35	-0.547812-2	0.348918-1
	301	-0.560853-2	0.354204-1
1767	302	-0.573894-2	0.359490-1
	303	-0.571817-2	0.364637-1
2630	306	-0.496595-2	0.394801-1
2631	307	-0.489579-2	0.391382-1
2632	308	-0.482089-2	0.389498-1
1793	309	-0.485118-2	0.384551-1
1794	310	-0.508261-2	0.378762-1
1795	311	-0.531581-2	0.374731-1
	312	-0.550661-2	0.372257-1
1796	313	-0.569740-2	0.369783-1

BY DBP DATE 2/1/79 SUBJECT Driver Vessel Inlet End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

First Thread - REV. 4 Design

NODE		DISPLACEMENTS	
Overall Model	Detail Model	δ_x (in.)	δ_y (in.)
1727	1	-0.404751-2	0.316656-1
	2	-0.425357-2	0.319419-1
1729	3	-0.445963-2	0.322182-1
	4	-0.457882-2	0.323892-1
1730	5	-0.469800-2	0.325601-1
	6	-0.487918-2	0.329522-1
1731	7	-0.506036-2	0.333443-1
	21	-0.546544-2	0.339096-1
1749	35	-0.587051-2	0.344748-1
	301	-0.613770-2	0.350358-1
1767	302	-0.640488-2	0.355968-1
	303	-0.656920-2	0.361526-1
2630	306	-0.592795-2	0.371501-1
2631	307	-0.590924-2	0.370064-1
2632	308	-0.588195-2	0.369870-1
1793	309	-0.593789-2	0.368854-1
1794	310	-0.614660-2	0.367605-1
1795	311	-0.636129-2	0.366967-1
	312	-0.654741-2	0.367026-1
1796	313	-0.673352-2	0.367084-1

ENGINEERING DESIGN & ANALYSIS SERVICES

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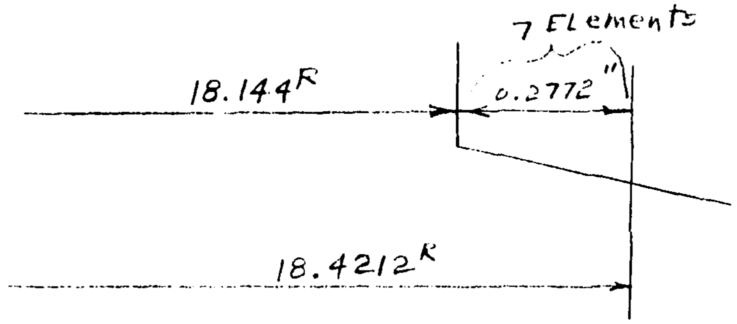
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 1/31/79 SUBJECT Driver Vessel Inlet End SHEET NO 1 OF 1

CHKD. BY DATE PROJ. NO JP1270

Pressure on 1st Thread - Original Design



Equivalent Thread Pressure - Original Design

Load = 378,073. Lbs/Radian

$$P = \frac{2(378{,}073.)\cdot\cos(7^\circ)}{\left[(18.4212)^2 - (18.144)^2\right]} = 74{,}044.91 \text{ psi}$$

$P = 0.1958481738$ (Thread Load)

Equivalent Thread Pressure - REV. 4 Design

Load = 85,503. Lbs/Radian

$P = 16{,}745.61$ psi

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 2/9/79 SUBJECT Driver Vessel Inlet End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Stresses in Driver Vessel Inlet End
Original Design - P = 60,000 psi

Thread No.	Thread Load (Lbs/Radian)	Stress Range (psi)
1	378,073.	286,574.*
2	390,925.	380,900.
8	243,857.	210,241.
9	228,027.	190,656.

* Maximum Surface Stress Intensity From Model with Elliptical Undercut.

Stresses in Driver Vessel Inlet End
REV. 4 Design - P = 60,000 psi

Thread No.	Thread Load (Lbs/Radian)	Stress Range (psi)
1	85,503.	165,999.*
2	132,189.	250,917.
8	313,093.	301,499.
9	313,492.	286,564.

* Maximum Surface Stress Intensity From Model with Elliptical Undercut.

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 12/13/78 SUBJECT Gas storage Vessel Inlet End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Equivalent Thread Pressures

DESIGN	THREAD NO.	Thread Load (lbs/Radian)	Thread Pressure (psi)
Original	9	228,027.	45,688.12972
REV. 2	2	102,971.8	20,631.71886
REV. 2	9	327,547.	65,628.23624
REV. 3	2	130,114.1	26,070.02627
REV. 3	9	315,672.	63,248.92791
REV. 4	2	132,189.05	26,485.76907
REV. 4	8	313,093.	62,732.19223
REV. 4	9	313,492.	62,812.13699
REV. 3	8	310,535.	62,219.66418
REV. 2	8	326,079.	65,334.10364
Original	7	259,016.	51,897.17274
REV. 1	7	326,650.	65,448.51081

$$P = \frac{2(\text{THREAD LOAD}) \cdot \cos(7^\circ)}{\left[(18.415)^2 - (18.144)^2\right]}$$

$$P = 0.2003628067\,(\text{THREAD LOAD})$$

Equivalent Pressure on 2nd Thread - Rev. 1 Design

Force on 2nd Thread = 101,411. Lbs/Radian

$$P_{Max} = \frac{2\,(101{,}411.)\cdot \cos(7^\circ)}{\left[(18.415)^2 - (18.144)^2\right]}$$

$$P_{Max} = 20{,}318.99259 \text{ psi}$$

Equivalent Pressure on 8th Thread - Rev. 1 Design

Force on 8th Thread = 330,358. Lbs/Radian

$$P_{Max} = \frac{2\,(330{,}358.)\cdot \cos(7^\circ)}{\left[(18.415)^2 - (18.144)^2\right]}$$

$$P_{Max} = 66{,}191.45609 \text{ psi}$$

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 11/22/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO. 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Equivalent Pressure on 4th Thread

Force on 4th Thread = 313,559. Lbs/Radian

$$P_{Max} = \frac{2(313,559.)\cdot \cos(7^\circ)}{\left[(18.415)^2 - (18.144)^2\right]}$$

$$P_{Max} = 62,825.5613 \text{ psi}$$

Equivalent Pressure on 8th Thread - original Design

Force on 8th Thread = 243,857. Lbs/Radian

$$P_{Max} = \frac{2(243,857.)\cdot \cos(7^\circ)}{\left[(18.415)^2 - (18.144)^2\right]}$$

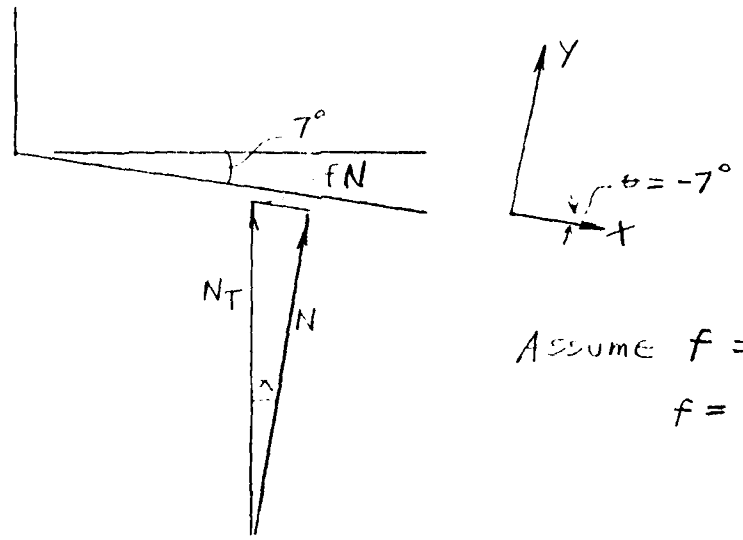
$$P_{Max} = 48,859.873 \text{ psi}$$

ENGINEERING DESIGN & ANALYSIS SERVICES — O'DONNELL & ASSOCIATES, INC. PITTSBURGH, PENNSYLVANIA

BY DBP DATE 11/20/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO. 1 OF 1

CHKD. BY ___ DATE ___ PROJ. NO JP1270

Friction Loading (2nd Thread - Original Design)



Assume $f = 0.122785$

$f = \tan\theta = \tan(7°)$

$N_T = (390,925)(\cos 7°) = 388,011$ Lbs/Radian

$N = (388,011)\cdot(\cos 7°) = 385,118.93$ Lbs/Radian

$fN = 47,286.659$ Lbs/radian $(f = 0.122785)$

Apply $FX = -C$ At Nodes 100 to 107 (8 Nodes)

$$C = \frac{47,286.659}{8} = 5,910.8324 \text{ Lbs/Radian}$$

$$P_{Max} = \frac{2(385,118.93)\cdot\cos(7°)}{\left[(18.415)^2 - (18.144)^2\right]} = 77,163.51 \text{ psi}$$

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 12/14/78 SUBJECT DRIVER VESSEL INLET END SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Friction Loading (8TH Thread - original Design)

$N_T = (243,857.)(\cos 7°) = 242,039.33$ Lbs/Radian

$N = N_T \cdot (\cos 7°) = 240,235.20$ Lbs/Radian

$fN = 29,497.174$ Lbs/rad $\{f = \tan 7°)$

$C = \frac{fN}{8} = 3,687.146731$ Lbs/Radian

$P_{Max} = 0.2003628067\,N = 48,134.19943$ psi

Friction Loading - 2nd Thread - Rev. 4 Design

$N = (132,189.05)\cdot[\cos^2(7°)] = 130,225.7601$ Lbs/Radian

$fN = 15,989.71278$ Lbs/Rad $\{f = \tan 7°)$

$C = \frac{fN}{8} = 1,998.714097$ Lbs/radian

$P_{Max} = 0.2003628067\,N = 26,092.39881$ psi

Friction Loading - 8th Thread - Rev. 4 Design

$N = (313,093.)\cdot[\cos^2(7°)] = 308,442.8999$ Lbs/Radian

$fN = 37,872.02603$ Lbs/radian $\{f = \tan 7°)$

$C = \frac{fN}{8} = 4,734.003254$ Lbs/Radian

$P_{Max} = 0.2003628067\,N = 61,800.48513$ psi

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 11/14/78 SUBJECT Gas Storage Vessel Inlet End Original Design SHEET NO 1 OF 1

CHKD. BY DATE PROJ. NO JP1270

Max. S.I. At Inlet End by Ratioing Outlet End Results by Forces

$$S.I.(Max) = \left(\frac{390,925}{448,381}\right)\left(\frac{60}{29.5}\right)(215,192) = 381,594 \text{ psi}$$

Versus 380,900 psi [illegible] % Difference

Factor For Inlet End for P = 47,500 psi

$$\text{Factor} = \left(\frac{60}{29.5}\right)\left(\frac{47{,}500}{60{,}000}\right) = 1.6101695$$

BY DBP DATE 12/15/78 SUBJECT Driver Vessel Inlet End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP/270

Rev. 4 Inlet End - With Friction

P (psi)	Factor
47,500	1.6101695
45,000	1.525423729
40,000	1.355932203
30,000	1.016949153
20,000	0.6779661017
10,000	0.3389830508
5,000	0.1694915254
0	0
25,000	0.8474576
15,000	0.5084746
26,000	0.8813559
24,000	0.8135593
22,000	0.7457627

$$\text{Factor} = \left(\frac{60}{29.5}\right)\left(\frac{P}{60,000}\right)$$

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
PITTSBURGH, PENNSYLVANIA

BY DBP DATE 12/15/78 SUBJECT Driver Vessel outlet End SHEET NO. 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Driver Vessel outlet End-with Friction

P (psi)	Factor
47,500	0.7916667
45,000	0.75
40,000	0.6666667
30,000	0.5
20,000	0.333333
10,000	0.1666667
5,000	0.0833333
0	0
25,000	0.4166667
15,000	0.25
26,000	0.4333333
24,000	0.4
22,000	0.3666667

$$\text{Factor} = \left(\frac{P}{60,000}\right)$$

ENGINEERING DESIGN & ANALYSIS SERVICES

O'DONNELL & ASSOCIATES, INC.
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BY DBP DATE 11/27/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 1 OF 1

CHKD. BY DATE PROJ. NO JP1270

Fatigue Life of Original Design
Inlet End – P = 60,000 psi

Thread No.	Stress Range, psi	Fatigue Design Life, N
2	380,900	130 cycles
8	210,241	693 cycles

Fatigue Life of Rev. 1 Design
Inlet End – P = 60,000 psi

Thread No.	Stress Range, psi	Fatigue Design Life, N
2	234,482	560 cycles
8	311,659	250 cycles

Fatigue Life of Original Design
Inlet End – P = 47,500 psi

Thread No.	Stress Range, psi	Fatigue Design Life, N
2	301,546	270 cycles
8	166,441	1,099 cycles

Fatigue Life of Rev. 1 Design
Inlet End – P = 47,500 psi

Thread No.	Stress Range, psi	Fatigue Design Life, N
2	185,632	888 cycles
8	246,730	497 cycles

BY DBP DATE 12/14/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Fatigue Life of Original Design
Inlet End – P = 60,000 psi

Thread No.	Stress Range, psi	Fatigue Design Life, N
2	380,900	130 cycles
7	228,994	581 Cycles

Fatigue Life of Rev. 1 Design
Inlet End – P = 60,000 psi

Thread No.	Stress Range, psi	Fatigue Design Life, N
2	234,482	560 cycles
7	326,650	208 Cycles

Fatigue Life of Original Design
Inlet End – P = 47,500 psi

Thread No.	Stress Range, psi	Fatigue Design Life, N
2	301,546	270 cycles
7	181,287	931 Cycles

Fatigue Life of Rev. 1 Design
Inlet End – P = 47,500 psi

Thread No.	Stress Range, psi	Fatigue Design Life, N
2	185,632	888 Cycles
7	259,645	446 Cycles

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BY DBP DATE 12/14/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

P = 60,000 psi
Thread Load and Stress Range for Original Design - Inlet End - Driver Vessel

Thread No.	Load (Lbs/Radian)	Stress Range (psi)
2	390,925.	380,900.
8	243,857.	210,241.
7	259,016.	228,994.

P = 60,000 psi
Thread Load and Stress Range for Rev. 1 Design - Inlet End - Driver Vessel

Thread No.	Load (Lbs/Radian)	Stress Range (psi)
2	101,411.	234,482.
8	330,358.	311,659.
7	326,650.	327,973.

BY DBP DATE 12/14/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 1 OF 2
CHKD. BY DATE PROJ. NO JP1270

Current Usage Factor For Inlet End of Driver Vessel

(a) Thread No. 2

$$K = \frac{380,900}{60,000} = 6.3483$$

$U_2^o = 0.177$ (From NSWC Curve) (See Appendix 5B)

(b) Thread No. 7

$$K = \frac{228,994}{60,000} = 3.8166$$

$U_7^o = 0.052$ (From NSWC Curve)

Cycles Remaining For Rev. 1 Design

$$U_2 = 0.177 + \frac{N_{II}}{888} \qquad \text{(Second Thread)}$$

$$U_7 = 0.052 + \frac{N_{II}}{446} \qquad \text{(Seventh Thread)}$$

(a) For $U_2 = 1.0$:

$$N_{II} = 888(1 - 0.177) = 731 \text{ cycles}$$

(b) For $U_7 = 1.0$:

$$N_{II} = 446(1 - 0.052) = 423 \text{ cycles}$$

The smallest value of N_{II} must be used.

Therefore, if the Rev. 1 Design is used the useful life remaining is 423 cycles.

Fatigue Life Remaining on Inlet End of Driver Vessel Based on P = 47,500 psi

Design	Critical Thread No.	Useful Life Remaining
Original	2	222 cycles
Rev. 1 Modification	7	423 cycles

$$N_{II}^{o} = 270(1-0.177) = 222 \text{ cycles} \quad \text{(Original Design)}$$

$$N_{II}^{1} = 446(1-0.052) = 423 \text{ cycles} \quad \text{(Rev. 1 Design)}$$

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BY DBP DATE 12/13/78 SUBJECT Gas Storage Vessel SHEET NO 1 OF 1
CHKD. BY DATE Inlet End PROJ. NO JP1270

Fatigue Life of Driver Vessel Inlet End
Original Design - P = 60,000 psi

Thread No.	Thread Load (Lbs/Radian)	Stress Range (psi)	Fatigue Design Life (Cycles)
2	390,925.	380,900.	133
9	228,027.	190,656.	842
8	243,857.	210,241.	693

Fatigue Life of Driver Vessel Inlet End
Rev. 2 Design - P = 60,000 psi

Thread No.	Thread Load (Lbs/Radian)	Stress Range (psi)	Fatigue Design Life (Cycles)
2	102,971.8	234,372.	554
9	327,547.	304,268.	263
8	326,079.	322,430.	219

Fatigue Life of Driver Vessel Inlet End
Rev. 3 Design - P = 60,000 psi

Thread No.	Thread Load (Lbs/Radian)	Stress Range (psi)	Fatigue Design Life (Cycles)
2	130,114.1	259,421.	482
9	315,672.	293,116.	296
8	310,535.	307,344.	255

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CHKD. BY DATE PROJ. NO JP1270

Fatigue Life of Driver Vessel Inlet End
Original Design – P = 47,500 psi

Thread No.	Stress Range (psi)	Fatigue Design Life (cycles)
2	301,546.	270
9	150,936.	1,328
8	166,441.	1,099

Fatigue Life of Driver Vessel Inlet End
Rev. 2 Design – P = 47,500 psi

Thread No.	Stress Range (psi)	Fatigue Design Life (cycles)
2	185,545	889
9	240,879	523
8	255,257	462

Fatigue Life of Driver Vessel Inlet End
Rev. 3 Design – P = 47,500 psi

Thread No.	Stress Range (psi)	Fatigue Design Life (cycles)
2	198,250	779
9	232,050	565
8	243,314	512

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BY DBP DATE 12/13/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 1 OF 2
CHKD. BY ____ DATE ____ PROJ. NO JP1270

Current Usage Factor For Driver Vessel Inlet End

(a) Thread No. 2

$$K = \frac{380,900}{60,000} = 6.3483$$

$$U_2^\circ = 0.177 \quad \{\text{From NSWC Curve})$$

(b) Thread No. 8

$$K = \frac{210,241}{60,000} = 3.5040$$

$$U_8^\circ = 0.033 \quad \{\text{From NSWC Curve})$$

Cycles Remaining For Rev. 2 Design

$$U_2 = 0.177 + \frac{N_R}{889} \qquad (\text{Second Thread})$$

$$U_9 = 0.033 + \frac{N_R}{462} \qquad \{\text{Eighth Thread})$$

By setting U_2 and U_8 Equal to 1.0, N_R For Each Thread is determined:

(a) For 2nd Thread:

$$N_R = 889(1-0.177) \doteq 732 \text{ Cycles}$$

(b) For 8th Thread:

$$N_R = 462(1-0.033) = 447 \text{ cycles}$$

The Smallest Value of N_R must be used.

Therefore, the Cycles Remaining for the Rev. 2 Design is 447.

BY DBP DATE 12/13/78 SUBJECT Gas storage Vessel Inlet End SHEET NO 2 OF 2
CHKD. BY DATE PROJ. NO JP1270

Cycles Remaining For Rev. 3 Design

$$U_2 = 0.177 + \frac{N_R}{779} \quad \text{(Second Thread)}$$

$$U_8 = 0.033 + \frac{N_R}{512} \quad \text{(Eighth Thread)}$$

By setting U_2 and U_8 Equal to 1.0, N_R For Each Thread is Determined:

(a) For 2nd Thread:

$$N_R = 779(1-0.177) = 641 \text{ cycles}$$

(b) For 8th Thread:

$$N_R = 512(1-0.033) = 495 \text{ cycles}$$

The smallest Value of N_R must be used.

Therefore, $N_R = 495$ cycles For the Rev. 3 Design.

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BY DBP DATE 12/9/78 SUBJECT Gas storage Vessel Inlet End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Fatigue Life of Driver Vessel Inlet End
Original Design – P = 60,000 psi

Thread No.	Thread Load (Lbs/Radian)	Stress Range (psi)	Fatigue Design Life (Cycles)
2	390,925.	380,900.	133
8	243,857.	210,241	693
9	228,027.	190,656.	842

Fatigue Life of Driver Vessel Inlet End
Rev. 4 Design – P = 60,000 psi

Thread No.	Thread Load (Lbs/Radian)	Stress Range (psi)	Fatigue Design Life (cycles)
2	132,189.	250,917.	480
8	313,093.	301,499.	271
9	313,492.	286,564.	319

BY DBP DATE 12/9/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Fatigue Life of Driver Vessel Inlet End
Original Design P = 47,500 psi

Thread No.	Stress Range (psi)	Fatigue Design Life (Cycles)
2	301,546.	270
8	166,441.	1,099
9	150,936.	1,328

Fatigue Life of Driver Vessel Inlet End
Rev. 4 Design – P = 47,500 psi

Thread No.	Stress Range (psi)	Fatigue Design Life (cycles)
2	198,643.	776
8	238,687.	533
9	226,863.	593

BY DBP DATE 12/9/78 SUBJECT Gas Storage Vessel Inlet End SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Cycles Remaining For Rev. 4 Design

$$U_2 = 0.177 + \frac{N_R}{776} \qquad \text{(Second Thread)}$$

$$U_8 = 0.033 + \frac{N_R}{533} \qquad \text{(Eighth Thread)}$$

By Setting U_2 and U_8 Equal to 1.0, N_R For Each Thread is Determined:

(a) For 2nd Thread:

$$N_R = 776\,(1-0.177) = 639 \text{ Cycles}$$

(b) For 8th Thread:

$$N_R = 533\,(1-0.033) = 515 \text{ Cycles}$$

The Smallest Value of N_R Must be Used.

Therefore, $N_R = 515$ Cycles For the Rev. 4 Design.

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BY DBP DATE 12/15/78 SUBJECT Driver Vessel Inlet End SHEET NO 1 OF 1
CHKD BY DATE PROJ. NO JP1270

Fatigue Life of Driver Vessel Inlet End
Original Design with Friction - P=60,000 psi

Thread No.	Thread Load (lbs/Radian)	Stress Range (psi)	Fatigue Design Life (cycles)
2	385,118.9	423,060	100
8	240,235.2	237,335	540

Fatigue Life of Driver Vessel Inlet End
REV. 4 Design with Friction - P=60,000 psi

Thread No.	Thread Load (lbs/Radian)	Stress Range (psi)	Fatigue Design Life (cycles)
2	130,225.76	256,546	457
8	308,442.9	335,265	195

BY DBP DATE 12/15/78 SUBJECT DRIVER VESSEL INLET END SHEET NO 1 OF 1
CHKD. BY DATE PROJ. NO JP1270

Fatique Life of Driver Vessel Inlet End
Original Design with Friction - P=47,500 psi

Thread No.	Stress Range (psi)	Fatique Design Life (cycles)
2	334,923	195
8	187,890	867

Fatique Life of Driver Vessel Inlet End
REV. 4 Design with Friction - P=47,500 psi

Thread No.	Stress Range (psi)	Fatique Design Life (cycles)
2	203,099	743
8	265,418	414

Cycles Remaining For Rev. 4 Design - with Friction

$$U_2 = 0.222 + \frac{N_R}{743} \quad \text{(Second Thread)}$$

$$U_8 = 0.06 + \frac{N_R}{414} \quad \text{(Eighth Thread)}$$

By setting U_2 and U_8 Equal to 1.0, N_R For Each Thread is Determined:

(a) For 2nd Thread

$$N_R = 743(1-0.222) = 578 \text{ cycles}$$

(b) For 8th Thread

$$N_R = 414(1-0.06) = 389 \text{ cycles}$$

The smallest Value of N_R must be used.

Therefore, $N_R = 389$ cycles for the Rev. 4 Design based on $P = 47{,}500$ psi.

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BY DBP DATE 12/15/78 SUBJECT Gas storage Vessel Inlet End SHEET NO 1 OF 1
CHKD. BY ___ DATE ___ PROJ. NO JP1270

SUMMARY of
Fatigue Life Remaining on Driver Vessel
Inlet End Based on P = 47,500 psi

Design	Critical Thread No.	N_R, Useful Life* Remaining	N_R with Friction
Original	2	222 Cycles	152 Cycles
Rev. 1	7	423 Cycles	
Rev. 2	8	447 Cycles	
Rev. 3	8	495 Cycles	
Rev. 4	8	515 Cycles	389 Cycles

* No Friction

$N_R^0 = 270(1-0.177) = 222$ cycles {Original Design)

$N_R^1 = 446(1-0.052) = 423$ cycles {Rev. 1 Design)

$N_R^2 = 462(1-0.033) = 447$ cycles {Rev. 2 Design)

$N_R^3 = 512(1-0.033) = 495$ cycles {Rev. 3 Design)

$N_R^4 = 533(1-0.033) = 515$ cycles {Rev. 4 Design)

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BY DBP DATE 12/18/78 SUBJECT DRIVER VESSEL INLET END SHEET NO 1 OF 1

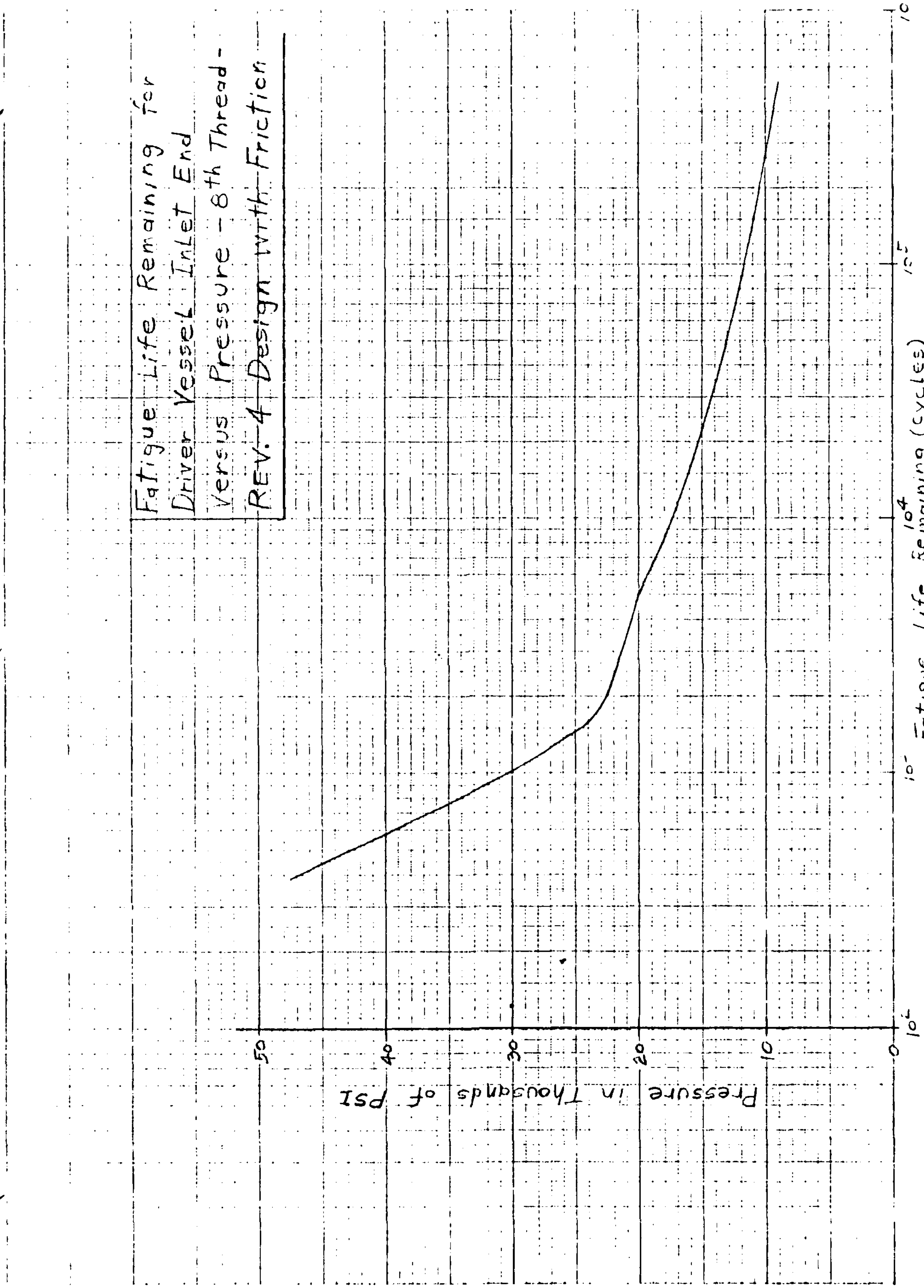
CHKD. BY DATE PROJ. NO JP1270

Fatigue Life of Driver Vessel Inlet End Vs. P
- 8th Thread - Rev. 4 - With Friction

P (psi)	Fatigue Life (cycles)	Fatigue Life Remaining (cycles)
47,500	414	389
45,000	478	449
40,000	611	574
30,000	1,084	1,019
25,000	1,543	1,450
20,000	5,376	5,053
15,000	24,171	22,721
10,000	278,066	261,382
26,000	1,430	1,344
24,000	1,669	1,569
22,000	2,646	2,487

N_R = Fatigue Life Remaining

$N_R = 0.94$ (Fatigue Life)


Fatigue Life Remaining for
Driver Vessel Inlet End
Versus Pressure - 8th Thread -
REV. 4 Design with Friction
Fatigue Life Remaining (cycles)
10^6
10^5
10^4
10^3
10^2
Pressure in Thousands of PSI
50
40
30
20
10
0

Inlet End - Rev. 4 Design - with Friction - P = 45,000 psi

If $\sigma = \Delta\sigma = 251,449$ psi and $K_{IC} = 100$ Ksi$\sqrt{in}$

1. $K_{IC} = 100$ Ksi$\sqrt{in}$

2. Critical Crack Depth

$$a_{cr} = \frac{1}{1.25\,\pi}\left(\frac{100,000}{251,449}\right)^2 = 0.040275''$$

3. Cycles to Failure

$C_o = 1.17366 \times 10^{-15}$ for ΔK in psi$\sqrt{in}$

$(n-2) = 0.25 \qquad M^{n/2} = (1.25\pi)^{1.125} = 4.659264564$

$\Delta\sigma^n = (251,449)^{2.25} = 1.415833891 \times 10^{12}$

$$\frac{1}{a_{cr}^{(n-2)/2}} = \frac{1}{(0.040275)^{0.125}} = 1.494066623$$

$$N = 1,033.280226\left[\frac{1}{a_i^{0.125}} - 1.49406\right]$$

$$a_i = \left(\frac{1,033.280226}{N + 1,543.789454}\right)^8$$

Inlet End - 8th Thread - Rev. 4 Design - with Friction
For $P = 45{,}000$ psi

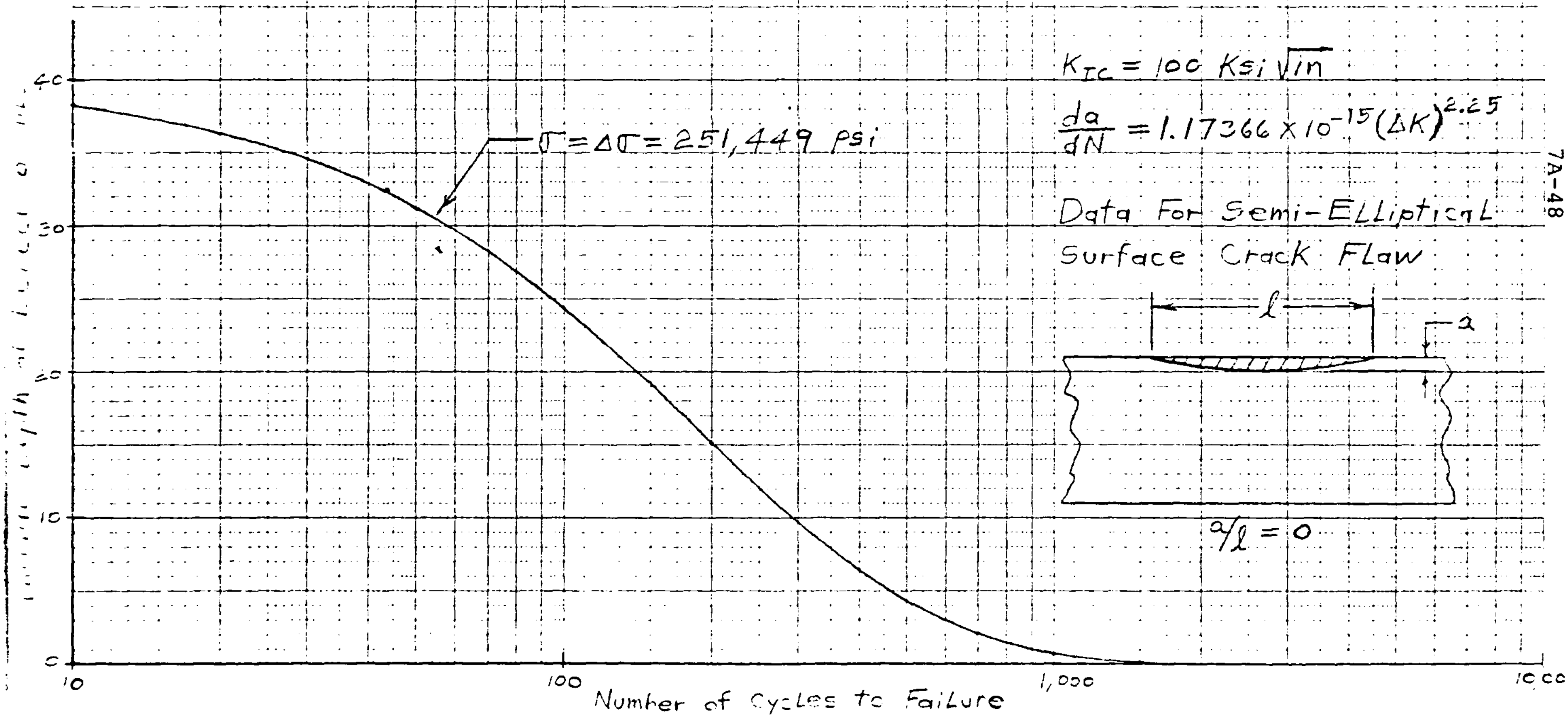
a_i Versus N for Threads
on Inlet End Closure
$\sigma = \Delta\sigma = 251{,}449$ psi, $K_{IC} = 100$ Ksi$\sqrt{in}$
Modified AISI 4340 Material

a_i inches	N Cycles
0.0382479	10
0.0363345	20
0.0312103	50
0.0282468	70
0.0243767	100
0.0151982	200
0.00972869	300
0.00637604	400
0.004268308	500
0.002022498	700
0.0007411321	1000
0.0000522398	2000
0.00000071515	3000

$$a_i = \left(\frac{1{,}033.280226}{N + 1{,}543.789454} \right)^8$$

FRACTURE MECHANICS EVALUATION OF DRIVER VESSEL INLET END

Initial Defect Size Versus Cycles to Failure For Inlet End - 8th Thread - REV. 4 DESIGN with Friction For $P = 45{,}000$ psi



APPENDIX 8A

PERIODIC INSPECTION
OF CRITICAL AREAS

PERIODIC INSPECTION OF CRITICAL AREAS

This Appendix contains a discussion of our recommendations for periodic inspection of the subject components.

Periodic inspection of critical areas of the driver and heater vessels will increase confidence that no flaws near critical size are present.

Analysis of critical areas on each vessel has shown the number of cycles to failure starting from a given flaw size. This size is the depth of a full circular crack around the vessel, and the number of cycles to failure is obviously a conservative, limiting value.

Discussions were held with C. Hellier and M. Bath of Nondestructive Test Engineering Division of Hartford Steam Boiler Inspection and Insurance Company. The discussions revealed the following sensitivities of liquid penetrant inspection techniques.

Type	Sensitivity*	
	Width	Depth
Zyglo ZL-15 high sensitivity water-washable liquid penetrant or other Group 1 or Group 6 penetrant per NAVASHIPS 250-1500	1-2 microns	20 microns
Magnetic Particle with AC device or DC Parker-Probe	1-2 microns	10 microns

*1 micron ≐ 0.0004 inch

Thus even a water-washable penetrant can reveal an 8 mil deep crack. For conservatism, it is reasonable to claim a sensitivity of 15 mils.

The recommended inspection frequency is based upon the philosophy of assuming the presence of an initial flaw depth of

15 mils. The possibility is accepted that a defect reaches the 15 mil limit of sensitivity immediately following an inspection. For added conservatism it is assumed that this defect is not found during the following inspection.

Since the units experience a variety of magnitudes of pressure cycles, it is desirable to account for the difference in crack propagation rates. To accomplish this it must be assumed that starting from any given point in time all future pressure cycles will be over the maximum pressure range. However, up to that point in time the affect of lesser magnitudes of pressure cycles can be considered.

To account for the possibility of not discovering an existing defect and to provide for sufficient remaining cycles once the defect is discovered on a subsequent inspection, a defect size, a_*, between the initial size of 15 mils (a_i) and the critical size (a_{cr}) is defined by the following*:

$$\frac{2}{3}\left(\frac{1}{a_i^{\frac{n-2}{2}}} - \frac{1}{a_{cr}^{\frac{n-2}{2}}}\right) = \left(\frac{1}{a_*^{\frac{n-2}{2}}} - \frac{1}{a_{cr}^{\frac{n-2}{2}}}\right) \qquad (1)$$

Starting from this defect size, i.e., a_*, two-thirds of the cycles required to generate the critical crack size from 15 mils with full range pressure cycling remains.

To reach the defect size a_*, the actual magnitudes of pressure cycling is considered. The crack growth rate is given by

$$\frac{da}{dN}^{\dagger} = C_o \, \Delta K^n / (1 - R)^{0.5} \qquad (2)$$

---

*See Appendix 5C for the basic equations and assumptions for crack propagation analysis.

†This form of the predicted crack growth takes into account the effect of mean stress. Reference "Fracture and Fatigue Control in Structures", Rolfe, S.T., and Barsom, J.M., Prentice-Hall, Inc., 1977, p. 248.

$$\frac{da}{dN} = C_o[\Delta\sigma a^{\frac{1}{2}} M^{\frac{1}{2}}]^n/[1 - R]^{0.5} \tag{3}$$

where R is the ratio of P_{min}/P_{max} and $(1 - R)$ equals $\Delta P/P_{max}$.

For numerical integrations let

$a = a_*$ on R.H.S. of equation (R.H.S. = Right Hand Side)

and

$$\Delta\sigma = \sigma_{ref}\left(\frac{\Delta P}{P_{ref}}\right)$$

Therefore

$$\Delta a = a_* - a_i = \sum_i C_o\, a_*^{n/2}\, M^{n/2}\left(\frac{\sigma_{ref}}{P_{ref}}\right)^n \Delta P_i^n/\left(\frac{\Delta P_i}{P_{max,i}}\right)^{0.5} \tag{4}$$

The crack size a_* will be reached and inspection is required when

$$\sum\frac{\Delta P_i^{n-0.5} P_{max,i}^{0.5}}{P_{ref}^n} = \frac{a_* - a_i}{C_o a_*^{n/2} M^{n/2}}\left(\frac{1}{\sigma_{ref}}\right)^n \tag{5}$$

Table 1 indicates the values used in the above equations. Also shown are the number of full pressure cycles required to extend a 15 mil defect to the critical size. The period between inspection might be extended slightly if in Equation (3), the average of a_i and a_* is used on the R.H.S. This results in about a 20% to 30% increase in $\Sigma\Delta P_i^n$.

Table 1

	$\frac{ref}{P_{ref}}$	a_{cr}, in.	a_*, in.	$\Sigma \frac{\Delta P_i^{n-0.5} P_{max}^{0.5}}{P_{ref}^{n}}$	Full Pressure Cycles to Failure	Reference
[illegible]	$\frac{223,510}{40,000}$	0.0510	0.0221	87	322	Appendix 7A P. 7A-46
[illegible]	$\frac{146,660}{22,000}$	0.1184	0.0282	317	1337	Appendix 6A P. 6A-38
[illegible]	$\frac{183,766}{12,000}$	0.0754	0.0248	164	646	Appendix 2C P.2C-6

.015"

1.1737(10^{-15})

= 1.25

= 2.25

*Based on 15 mil initial flaw depth.

APPENDIX 9

THERMAL CONSIDERATIONS

Thermal Considerations

During operation, the working gas temperature varies which in turn produces thermal stresses within the steel components. Lacking specified gas temperature transients the effect of thermal transients can be given only speculative attention.

In general, the transient thermal effects are judged to be insignificant in affecting the predicted cyclic life of the components. The reasoning is that the gas flow period is very short (on the order of 1 or 2 seconds) relative to the diffusivity and thickness parameters of the involved components. In addition, the relatively small thermal capacitance of the gas results in a rapid attainment of thermal equilibrium following the flow period such that little change occurs in the component average temperatures. The thermal stress responses are most probably "skin" type stresses at the gas boundaries.

The most limiting locations are generally at the threads of the various threaded closures which are in themselves not gas boundaries. These locations see insignificant if any, thermal stresses and, therefore, the temperature transients probably have little affect on cyclic life.

For other locations of the components, thermal stresses may be more dominant than the primary pressure stresses. In the MACH 10 heater, prior to flow initiation, steady state thermal gradients may produce significant stresses if the vessel is redundantly supported. If the heater is not redundantly supported bowing will occur and the thermal stresses are probably insignificant but reactions may then cause significant stresses in the connecting hardware.

The complexities of the system and the thermodynamics of the operations preclude a purely analytical approach to determine the gas temperatures required to evaluate metal temperatures. A more definitive evaluation of thermal effects would require a combination of extensive temperature measurements and analysis.